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SEATTLE, WASHINGTON

DEVELOPMENT AND TESTING OF A LOW-TEMPERATURE HYDRAULIC SYSTEM

Final Report

Volume I

ACTUATION APPLICATIONS TRADE STUDY

D2-90795-1

Prepared under

George C. Marshall Space Flight Center Contract NAS8-11722

by

The Boeing Company, Aerospace Group

September 1965

CONTENTS

FIGURES

ABSTRACT

This document is Volume I of the final report for Contract NAS8-11722. Abstracts for both volumes are included below for continuity.

Volume I — The stages of space exploration leading to placement of men on the Moon require Earth-orbiting vehicles, with and without re-entry capability, and lunar surface vehicles, with and without roving capability. Actuation tasks must be performed during operation of each of these four types of vehicles. The comparison of various actuation systems to satisfy these space tasks is determined in this study by the use of a parametric trade evaluation.

Eighty candidate space actuation tasks are considered, from which 10 are selected as having the most trade-study value. From these 10, one task associated with each of the above four vehicle types is investigated in detail.

The evaluation considers hydraulic, pneumatic, and electromechanical actuation systems to satisfy the requirements of each task. Provisions for power and thermal conditioning for each system are considered. The results of the trade study are in the form of a numerical rating summary showing the comparison of systems, for the same task, on the basis of weight, reliability, cost, availability, and performance margin.

Volume II — The effective use of hydraulic systems in space applications depends mainly on the compatibility of these systems with space temperature and pressure environments. The results of testing some of the more important compatibility aspects are presented in this report. Advanced fluids and seal compounds are evaluated to provide a basis for the selection of these items for a hydraulic system operating in a space thermal environment of -240 to +275°F and a hard vacuum of 10^{-8} torr. Strength properties of metals are analyzed to select materials for component fabrication. A single-pass linear-actuator test system and supporting equipment are designed to simulate the operational characteristics of a study task evaluated as described in Volume I.

Results of the test program show how reducing the fluid temperature affects the operation of the system and the minimum operating temperature for the test fluid selected. The results of operation in a simulated space flight show the system's capability for intermittent use associated with many space actuation duty cycles. The effects of high temperature on system operation are reported, as are the effects of 100 hours of system thermal cycling at temperatures from -240 to +275°F. Environmental effects on seals and system operation following long-term (5-month) hard-vacuum storage of the linear actuator are reported. A thermal analysis is also presented that describes, using parametric data, the protection required for various hydraulic applications in space.

1.0 INTRODUCTION

The space vehicles launched to date for orbital or space missions have required relatively simple actuation systems of low power output. The Ranger vehicle employs most of the common types. Solar panels are extended by a mechanical system, the high-gain antenna is operated by an electromechanical system, the gamma-ray boom by a pneumatic system; and a radiation-measuring boom by a hydraulically damped system. Similar tasks are performed by actuation systems on the majority of existing space vehicles.

The evolution from unmanned satellites to manned space vehicles has been accompanied by many new actuation tests. Manned space vehicles now under development or being studied for missions of long duration (vehicles such as Apollo, LEM, MORL, AORL, and MOL) suggest more complicated tasks and additional uses for power-amplifying actuation systems. The increased involvement of man in space travel has expanded the size of the space vehicle and increased the magnitude of subsystem support so that greater use of powered actuation is necessary. The determination of the extent to which powered actuation systems must be used on manned space vehicles is a prerequisite to the development of such systems. Extremely long or intermittent use in space will require different design approaches.

The type of actuation system that most adequately satisfies the requirements of these new space actuation tasks must be determined to orient and concentrate system development. Such a determination might be accomplished by a trade study that analyzes the desirable features of each type of actuation system when applied to the space vehicle tasks for which it is most competitive.

The need for this evaluation of space actuation systems is emphasized by the timetable of planned interplanetary travel in the 1970's using vehicles that will require such systems. The program reported in this document is a significant step toward this goal.

2.0 SUMMARY

The involvement of man in space exploration is accompanied by an increased need for supporting subsystems including actuation systems. The choice of actuation concept is not always obvious. An actuation trade study is provided in this report in which about 80 tasks are considered as candidates to show the comparison between hydraulic, pneumatic, electromechanical, and mechanical actuation concepts. These include the majority of the actuation tasks presently visualized as being required on future space vehicles. Each task is associated with one of four manned space vehicle categories. These categories are: (1) an orbiting vehicle with subsequent re-entry, (2) an orbiting vehicle without re-entry, (3) a lunar landing vehicle, and (4) a lunar roving vehicle.

Actuation requirements are developed for 10 tasks from the above 80 that have the most significant trade study value. Power levels for these 10 tasks range from 0.2 horsepower to 3600 horsepower with actuation rates from 2 inches per second to 15 feet per second. Three of the 10 tasks involve flight control functions. A detailed trade study is performed to evaluate four tasks, selected from the 10, which are considered to provide the best trade data for all the listed actuation methods. One task is selected for evaluation in each of the vehicle categories.

The first task evaluated is the elevon surface actuation on an orbiting vehicle with re-entry capability. This task requires 7 horsepower to develop a maximum torque of 144,000 inch-pounds at the load with a rate of 15 degrees per second. The thermal environment is -60 to +550°F; therefore, cooling systems are required for some of the actuation techniques.

The task selected on an orbiting vehicle without re-entry capability is that of a centrifuge drive. This drive is to provide an environment between 0.01 and 6.0 g's for 45 minutes each day. Actuation equipment is located in the crew ambient thermal environment; nevertheless cooling is required because of the length of operation time. A pneumatic method of actuation is not practical for this task because of the problem in providing a long-duration pneumatic power source.

The thrust vector control actuation during midcourse and terminal maneuvers for a lunar landing vehicle is the third task evaluated in the trade study. This operation requires 2 horsepower to gimbal an RL-10 engine ±7.5 degrees at a maximum rate of 20 degrees per second. The thermal environment is -240 to +100°F. Heating is required for the hydraulic concept.

The fourth task is the payloader actuation of a materials handling device attached to a lunar roving vehicle. The payloader requires 187,000 inch-pounds of torque to pick up and elevate 1 cubic yard of lunar dust for caisson coverage; the shelter is 40 feet high. Twelve hundred cycles of operation are required per shelter. The temperature on the lunar surface is -240 to +275°F; both heating and cooling must be provided for actuation equipment.

Actuation system preliminary designs are prepared for each space task using, where possible, suggestions from component suppliers on available actuation equipment. The elevon and TVC designs for all concepts incorporate redundancy in critical components emphasizing the importance of reliability where flight safety is involved. If necessary, each actuation system design is supplemented by a thermal conditioning system design and an auxiliary power system design. Advantage is taken of power that could be made available from the basic vehicle without penalty to the actuation systems. Mechanical concepts were not feasible except in the case of the payloader, and this system bore such close resemblance to the electromechanical system that evaluation was not undertaken. The payloader task requires dual actuation systems because bucket actuation at the end of the extending 40-foot boom cannot be centrally powered.

The comparative evaluation of the actuation system designs is conducted using the parameters of weight, reliability, cost, availability, and performance. Weights are calculated for each evaluated system based on available actuation components. Because hardware suggested for the actuation designs has not been used in space systems, a true measure of reliability, cost, availability, or performance has not been established. A numerical representation showing the competitive positions of components in relation to these four parameters is developed based on experience with similar components. Each actuation system is given an overall rating that is a combined expression of the ratings for individual parameters. This overall rating reflects the differences in the degree of importance of each evaluation parameter with respect to each vehicle mission.

The results of the comparative evaluation using the numerical rating system show that the pneumatic TVC actuation system is distinctly superior to the other systems. Similarly the electromechanical centrifuge actuation system is significantly superior. The evaluation of the remaining two tasks — elevon control and payloader actuation — showed no distinct superiority for one actuation concept; therefore, hydraulic systems are recommended on the basis of extensive operational experience at these power levels.

The trade study results show that each of the hydraulic, pneumatic, and electromechanical concepts has application to space tasks. Because only four were selected for study, many tasks for which hydraulics might be recommended have not been analyzed. The reservation to use hydraulics for actuation tasks in the severe environments of space has been clarified to a degree by the trade study. The evaluations show that all space actuation systems need not operate in a temperature environment considered severe for hydraulics and, even in the severe environment, environmental conditioning penalties are less for hydraulics than similar penalties for some of the other systems. The testing conducted as part of this contract and reported in Volume II emphasizes that hydraulic operational problems at low temperature are not as severe as originally anticipated. The development and evaluation of advanced low-temperature hydraulic fluids has influenced their acceptance in other than actuation systems; an example is the environmental control systems for space vehicles. The confidence provided by extensive development and experience with conventional and advanced hydraulic systems is therefore considered an important factor in the selection of space vehicle actuation systems where human safety is involved.

3.0 APPLICATIONS SELECTION

Man's occupation of large orbiting spacecraft or lunar roving vehicles, in contrast to unmanned satellites, requires vehicle subsystems to support man, additional actuation tasks to perform the mission and service the vehicle, and increased subsystem reliability to guarantee man's safety. The significance of these requirements and the proper selection of actuation methods to perform these tasks will increase as power levels increase and man becomes more involved with controlling space vehicles.

The purpose of this study is to determine by parametric evaluation the relative comparison of hydraulic, pneumatic, electromechanical, and mechanical actuation methods for selected future space actuation tasks.

3.1 ACTUATION TASK SURVEY

Figure 1 shows actuation tasks for present and planned spacecraft missions as advanced as lunar exploration; Mars missions are not included. The tasks are categorized by their association with particular vehicle groupings and missions. The first group consists of vehicles with Earth-orbiting capability with subsequent Earth atmosphere re-entry. This group is typified by the 12-man shuttle vehicles such as the M-2 or HL-10 configurations. The second group of vehicles consists of those with orbiting capability but no re-entry capability. Many unmanned vehicles are in this category; however, vehicles of most interest to this study are the manned space stations similar to MOL and larger stations.

The third group includes vehicles with lunar landing capability, but no roving capability on the lunar surface. Typical vehicles are lunar base designs and LEM. The fourth group consists of lunar surface operational vehicles. Numerous jobs will be performed by such vehicles, and it is considered that the most versatile design would be a multipurpose roving vehicle with numerous power attachments for specific applications.

3.2 COMPETITIVE TASKS

The actuation tasks considered adaptable to these vehicles vary from antenna orientation at a fraction of a horsepower to re-entry control surface actuation greater than 10 horsepower. Not all tasks considered are adaptable to competitive evaluation between actuation concepts. The fractional-horsepower tasks are less competitive for evaluation than higher-power tasks — some involve special environments and others definitely favor electromechanical actuation methods. The limited number of high-powered actuation tasks (of more than about 10 horsepower) appear more favorable to hydraulics than to other actuation system concepts. For horsepower values between 1 and 10, it is not definite that one system is preferable over another, and a trade study is needed to select the optimum concept. For this reason, actuation tasks requiring 1 to 10 horsepower were selected for evaluation in this study.

Table 1 - Space Vehicle Actuation Tasks

Space Vehicle Group	Actuation Task	Duration	Type of Actuator	Characteristics
Group 1 Orbiting with subsequent re-entry capability	1. Re-entry Aerodynamic surface control	Continuous for approx 1 hour, possible long space storage	Hydraulic ~ Elect Mech	High resolution and response
	Wing sweep opening	Single actuation	Hydraulic	Locking
	Landing operation	Single actuation	Pneumatic	Locking
	Thrust vector control (TVC)	Few seconds during either injection or retrocession from orbit	Hydraulic ~ Elect Mech	High resolution and response
	2. Erection & Retraction of Collapsible-Type Mechanisms Radar antenna Solar collector for thermomechanical & thermoelectric power Space radiator Solar-cell panel	Single actuation Duration depends on the type of mechanism	Elect Mech ~ Mechanical ~ Comp gas ~ Hydraulic accumulator	Positioning and Locking
	3. Orientation of Equipment Solar-cell panel Solar collector Space radiator Target-seeking for military purpose Photo camera Television camera Tracking devices and telescopes Radar antenna	Intermittent during the orbital mission	Elect Mech~ Hyd if rotating power source is available onboard	Response and resolution depend on the functions
	4. Ejection & Separation Stage separation Ejection of expendable equipment and materials upon re-entry	Single operation Short duration	Solid charge Mechanical Compressed gas	Interference with trajectory must be prevented
	5. Miscellaneous Mechanisms Latch for locking and opening hatches during rendezvous	Single or repeated operation	Mechanical ~ Comp gas ~ Elect Mech ~ Pneumatic ~ Hydraulic	Depends on the functions
	Life support functions in crew cabin (artificial gravity, exercise mechines, etc.)	May be long		
Group 2 Orbiting without re-entry capability	The same as Group 1, except No. 1 If unmanned, No. 4 (ejection portion), and No. 5 not required			
Group 3 Space Travel & Landing	The same as Group 2, plus 6. TVC for midcourse correction, lunar orbital injection, landing brake propulsion and takeoff propulsion	Intermittent, short operational, duration, possible long storage period	Elect Mech ~ Hydraulic	High resolution and response
	7. Deployment of landing mechanism	Single operation, possible long space storage	Solid charge ~ Comp gas ~ Mechanical	Locking
	Shock absorber		Hyd accum ~ Liq spring ~ Pneumatic~ Mechanical	High shock load
Group 4 Lunar Surface Operation	1. Erection & Retraction of Mech 2. Surface Mod & Construction, Bulldozing, embankment, clearing, grading, erection 3. Materials handling 4. Functions for transportation (roving vehicles)	See Group 1, No. 2 Long, repeated operations similar to operations on Earth long space storage	Elec Mech ~ Hydraulic if rotating power source is available	Depends on the functions
	5. Life-support functions 6. Orientation 7. Miscellaneous mechanisms	See Group 1, No. 5 See Group 1, No. 3 See Group 1, No. 5		

Figure 1: Space Vehicle Actuation Tasks

The list of possible space actuation tasks, see Figure 1, was reduced to 10 candidate tasks judged best for competitive comparison of actuation concepts. Preliminary requirements were determined for each of these tasks. These requirements are listed in Figure 2, which also shows the association of each task to its vehicle type.

3.2.1 TASK 1 — ELEVON SURFACE CONTROL FOR RE-ENTRY SHUTTLE VEHICLE (References 1 and 2)

The elevon surface actuation on the 12-man shuttle version of the M-2 or HL-10 vehicle configurations provides pitch and roll control during atmospheric descent and landing of the vehicle. The two elevons must survive space storage of up to 6 months, possibly without thermal conditioning. The elevons are used only when the vehicle is traveling in a re-entry pattern from low earth orbit thereby withstanding only a 10-minute operational time. Attitude control that is required for space maneuvers is accomplished using reaction jets. The maximum elevon design hinge moment occurs when the elevons are used for escape during boost abort at high dynamic pressures (1300 psf) and a deflection angle of about 20 degrees.

3.2.2 TASK 2 — THRUST VECTOR CONTROL FOR SPACE PROPULSION MODULE (References 2 and 3)

Thrust vector control of an engine, such as the RL-10 in a space propulsion stage, is one of the actuation tasks chosen to represent orbiting vehicles without re-entry capability. TVC operation is confined to periods of main engine firing, which include: (1) a maximum of four midcourse velocity corrections of short duration (20 seconds maximum) during translunar coast; (2) two long-duration (about 300 seconds each) firings for injection into lunar orbit and injection into trans-Earth trajectory; and (3) four more midcourse corrections during the journey back to Earth. Before each firing, the vehicle is oriented to the proper attitude by the reaction control system and the gimbal angles are adjusted so that the thrust will be directed through the predicted location of the vehicle center of gravity. The vehicle attitude control loop is closed at all times so that unpredictable offsets, misalignments, and disturbances can be compensated by the thrust vector control system.

3.2.3 TASK 3 — CENTRIFUGE ACTUATION FOR LARGE MANNED SPACE STATION (Reference 4)

A centrifuge on board a large space station will supply various levels of centrifugal force to relieve any adverse physiological effects of zero-g environment and will condition the astronauts for re-entry loads. The centrifuge consists of dual counter-rotating assemblies of the maximum radius permitted by the space station design. Each of the two assemblies provides four personnel carriers that can be adjusted from a seated position to radially extended positions. Conditions from 0.01 to 6 g can be achieved. Actuation equipment will be within the controlled space station environment.

3.2.4 TASK 4 — HANGAR DOOR ACTUATION FOR LARGE MANNED SPACE STATIONS (Reference 5)

A large hangar door is required in the preliminary design of a large space station. The door provides docking access to the station for re-entry capsules used for the return or rotation of crew members. Actuators open and close the hangar door and absorb the kinetic energy of the door to prevent impact damage to the space vehicle structure and door. Door-opening loads are small; the snubbing requirements establish the design of the actuator. Door-latching actuators are also used. These actuators compress the O-ring seal on the door and lock the door so that the hangar can be pressurized for a shirt-sleeve environment.

3.2.5 TASK 5 — SOLAR CELL PANEL ERECTION AND ORIENTATION FOR LARGE MANNED SPACE STATION (Reference 4)

Large solar panels have been recommended for a major portion of the secondary power requirements of a long-duration, large space station. An erection actuation mechanism must move the solar panels from their stowed position to their deployed position. The mechanism must also keep the solar panels continuously facing the Sun as the space vehicle orbits the Earth. Two solar panels (about 50 by 100 feet each) must be erected through a 90-degree arc in 5 minutes. Each panel will weigh about 6850 pounds on Earth. The torque to move the panel in a zero-g environment is small (33 foot-pounds). The kinetic energy of each panel that must be absorbed at the end of erection is 53 foot-pounds. This actuation function is performed only once, actuator size is dictated by a 3-foot stroke requirement. The actuator will be located in an unpressurized area but will be protected by a fairing. The environment will approach -200 to +300°F.

Assuming that the space station has a 90-minute orbit around the Earth and that the solar panels need be oriented toward the Sun in only one plane, a continuous 4-degree-per-minute orientation is required. In a zero-g environment, only the friction of the mechanism must be overcome. It is assumed that this friction will slow the orientation of the panel 1-degree-per-minute. To overcome friction, the solar panel orientation mechanism must exert a continuous torque of 0.218 foot-pounds to keep the panels always facing the Sun.

3.2.6 TASK 6 — LANDING MECHANISM FOR LUNAR LANDING VEHICLE (Reference 6)

The landing mechanism of a lunar landing vehicle must engage the lunar surface, absorb the kinetic energy of the vehicle at landing, and support the vehicle. The gear must function on uneven ground, level the vehicle for takeoff, be reusable, and be able to be retracted for compact storage. A typical gear will consist of four self-aligning strut mechanisms that are equally spaced around the periphery of the vehicle. Actual contact with the lunar surface will be made by round pads attached to the struts. Each pad will actuate an energy-absorption mechanism. The stroke of the energy absorber is assumed to be 3 feet and the total energy required to be absorbed is 400,000 foot-pounds for a 17,000-pound lunar vehicle (100,000 pounds on Earth).

3.2.7 TASK 7 — THRUST VECTOR CONTROL FOR LUNAR LANDING VEHICLE (Reference 3)

A lunar landing vehicle thrust vector control system provides vehicle attitude control during periods of main propulsion system operation. This includes braking into equal-period lunar orbit, descent braking, hover, and landing. The thrust level is controllable for hovering while selecting the landing site, and fast response is required for rapid maneuvering just prior to touchdown. TVC requirements are based on the results of a past Boeing study that used an analog simulation of piloted lunar landing.

3.2.8 TASK 8 — MATERIALS HANDLING FOR LUNAR ROVING VEHICLE (References 7 through 10)

Materials handling actuation functions of a lunar roving vehicle are typified in the design of a vehicle with manipulators similar to a vehicle designed by General Electric. Such manipulators provide nine motions including hand, hook, wrist, elbow, upper arm, and shoulder. The requirements considered picking up 600 lunar pounds and an arm plus hand extension of 16 feet. The manipulator, designed for remote experiments on the lunar surface, will also be required during handling and repair of nuclear powerplants and general maintenance tasks.

3.2.9 TASK 9 — SURFACE MODIFICATION FOR LUNAR ROVING VEHICLE (References 7 and 9)

Lunar surface modification tasks include items such as excavating, embankment, clearing, grading, and loading. Many tasks provide meteoroid and radiation protection for lunar shelters. The surface modification vehicle (bucket loader) is assumed to have a bucket capacity of 1 cubic yard and an operational duty function of 300 to 1200 cycles to fill a caisson around a lunar shelter. Actuation tasks include: (1) boom extension, which extends the boom and forces the bucket through the soil until the bucket is full, (2) the boom elevation to raise and lower the boom, and (3) bucket actuation to position the bucket during filling and dumping. The elevation actuator requires the greatest horsepower and is the only load listed in Figure 2.

3.2.10 TASK 10 — MOTIVE POWER FOR LUNAR ROVING VEHICLE (References 7 through 9)

A typical lunar vehicle will have four to six powered wheels requiring a total of 6 horsepower; the vehicle will weigh approximately 6000 pounds (1000 pounds on the Moon). The range of the vehicle will be about 250 to 380 miles and the maximum speed will be approximately 11 mph on a nominal grade of 3-degree slope. Wheel speed will be about 60 rpm at 11 mph. Rotation of the wheels forward and backward, braking the wheels, steering the wheels by differential braking, and disconnecting each individual wheel from the wheel power unit are all actuation functions to be performed.

3.3 TASK SELECTION

One actuation task from each listing of tasks associated with a vehicle group was selected for detailed evaluation (refer to Figure 2). These were tasks considered most competitive for the four actuation concepts. Actuation tasks requiring less than 1 horsepower were not considered competitive for all concepts. Similarly one-shot actuation systems were not considered competitive so that a more realistic trade study could be performed by concentrating on continuously operating systems. The surface modification task using the lunar rover was selected in preference to the other lunar roving vehicle tasks because funded contract efforts have been awarded in the materials handling and motive power areas (see References 7 and 10).

The following four tasks were selected for detailed evaluation:

1) Elevon surface control (re-entry shuttle vehicle);
2) Centrifuge actuation (large manned space stations);
3) Thrust vector control (lunar landing vehicle);
4) Surface modification (lunar roving vehicle).

Figure 3 shows a general illustration of each actuation task.

Vehicle Type	Actuation Type	Maximum Movement	Maximum Rate	Type Operation	Break Frequency cps	Operating Time	Power hp	Load
Orbiting with Re-entry	<u>Re-entry Shuttle Vehicle</u> 1. Elevon-surface control*	+40° to - 30°	15°/sec	control + space storage	3.0	10 min	7	144,000 in.-lbs
Orbiting without Re-entry	<u>Space Propulsion Module</u> 2. Thrust-vector control	±7.5°	20°/sec	control + space storage	3.2	14.2 min	2	3,900 in.-lbs
	<u>Large Manned Space Station</u> 3. Centrifuge*	---	3.6 rad/sec	control	---	45 min/day	6	10,750 in.-lbs
	4. Hangar-door actuation	120°/opening	1.8°/sec	intermittent	---	66 sec/every 90 day	0.6	129,500 in.-lbs
	5. Solar-cell panel, erect/orient	90°/opening	0.3°/sec/open 4°/min/orient	control	---	5 min/open cont/orient	0.2	636 in.-lbs
Space Travel and Landing	<u>Lunar Landing Vehicle</u> 6. Landing mechanism	3 ft	15 ft/sec	intermittent + space storage	---	1 sec/landing	3600	17,000 lbs
	7. Thrust-vector control*	±7.5°	20°/sec	control + space storage	3.5	8 min	2	3,900 in.-lbs
Lunar Surface Operation	<u>Lunar Roving Vehicle</u> 8. Materials handling	18 in.	2 in./sec	control + space storage	---	---	8	11,500 in.-lbs
	9. Surface modification*	85 ft hor veh. 40 ft vert boom	50 sec/cycle	control + space storage	---	4 hrs/job	5	187,000 in.-lbs
	10. Motive power	380 miles	11 mph	control + space storage	---	40 hrs	6 total	1,000 in.-lbs per wheel 6 wheels

*Tasks on which trade studies were conducted.

Figure 2: Actuation Requirements

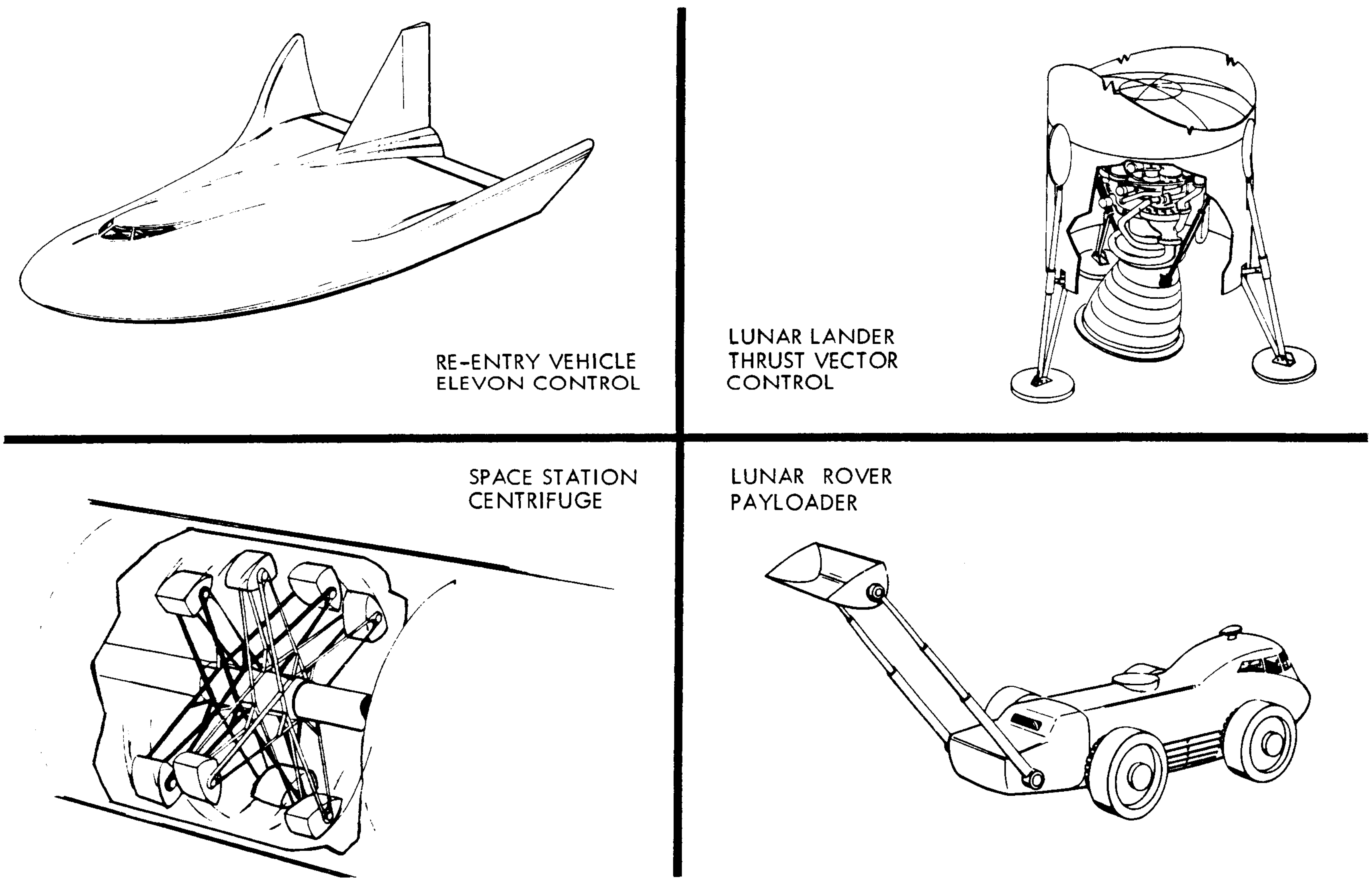


Figure 3: Selected Tasks

4.0 TASK REQUIREMENTS

The requirements of the selected tasks vary significantly. The power level for most of the tasks is low because response and load requirements have decreased with the advent of long mission times and the absence of aerodynamic influences. Long storage in space with intermittent usage for short durations is a characteristic of many tasks.

Requirements were developed for each task and used as ground rules for the preliminary designs of hydraulic, pneumatic, electromechanical, and mechanical actuation systems to operate the actuated devices.

4.1 ELEVON ACTUATION

Elevon surface actuation requirements were developed for one of two elevons on a 12-man shuttle vehicle. Flight actuation requirements were assumed to be similar to those presented in Reference 1. The most important requirements are tabulated in Figure 2.

The maximum load of 144,000 inch-pounds results from the design condition for vehicle abort during boost. During this period, the elevon would be rotated hard-over against a maximum dynamic pressure of 1300 psf and held there until the vehicle was clear of the booster. Flyback time during an abort would be less than the 10-minute re-entry operating period of a normal maneuver.

The maneuver load during re-entry will be 10 percent or less of the maximum load during 9.5 of the 10 minutes. The remaining 0.5 minute will consist of load spikes, each of about 3-second duration varying from 20 to 40 percent of maximum load and occurring in random order throughout the 10-minute maneuver.

The elevon actuation system will be environmentally controlled during periods when the shuttle vehicle is in flight. When the vehicle is docked at a space station, the equipment will be exposed to uncontrolled environment. Power will be available from the space station to warm the equipment before the return trip begins.

To provide a common basis between the preliminary designs for the various actuation concepts, a minimum actuator lever arm of 5 inches was selected. A second common design selection was to provide redundant flight control system components that were considered failure-sensitive. These items included such components as servovalves, pumps, clutches, and motors.

Power for the elevon actuation system can be considered to be provided from a central vehicle source that supplies all powered utilities on the shuttle vehicle. Penalty to the actuation system will be in the additional weight of fuel and equipment in the power system to provide the elevon power.

4.2 CENTRIFUGE ACTUATION

The centrifuge used on a manned orbiting space station is considered to be required 45 minutes per day to provide physiological conditioning of the crew. The centrifuge, consisting of two parallel counterrotating wheels of about 30-foot diameter, is considered to weigh 1000 Earth pounds. The actuation system drives the wheels to provide a maximum 6-g artificial gravity and a braking time of 30 seconds. Each wheel is capable of carrying four men; a two-man load is considered necessary to provide balance. The centrifuge compartment in which the actuation equipment is contained will be in a controlled environment. No redundancy of components is considered for this task because the entire actuation package can be easily replaced by the space station crew.

The power to operate the centrifuge can be considered to be available at no penalty to the actuation system. Such power is considered available because the magnitude is low and the men on the centrifuge are not operating equipment that draws power.

4.3 THRUST VECTOR CONTROL ACTUATION

The thrust vector control actuation system provides gimbaling power for an RL-10 propulsion engine. Reference 11 gives engine specifications. The actuation system must operate only during the 8 minutes of intermittent engine operation for braking into equal-period lunar orbit, descent braking, hover, and landing. Maximum engine deflection is ±7.5 degrees and requires an operating torque of 3900 inch-pounds. Engine trim deflection of about ±0.5 degree requires 2750 inch-pounds of torque. A reasonably optimized actuator mounting position was selected to provide similarity between actuation concept evaluations. This mounting requires a 4-inch actuator stroke, 6-inch-per-second velocity, and a 7-inch minimum lever arm. Redundancy for the thrust vector control system is the same as determined for the elevon system.

Thrust vector control actuation components are exposed to space environments throughout the vehicle flight. Before propulsion engine ignition, the engine will be facing away from the Sun to minimize boiloff of cryogenically stored liquids. In this vehicle attitude, gimbal actuation equipment is expected to reach a temperature of -240°F and must be warmed before use. In other vehicle attitudes the equipment is not expected to reach a conditioning temperature greater than +100°F.

Shaft horsepower, up to 2.5, is available from an engine turbopump pad without penalty to the system to operate the actuation equipment. Similarly hydrogen bleed at the inlet to the turbine is available with essentially no penalty to the system.

4.4 PAYLOADER ACTUATION

The payloader is considered an attachment to a lunar roving vehicle. Such attachments as the payloader, bulldozer, or boring machine are to be coupled to a central vehicle body having motive power. In this manner, the central vehicle would serve several purposes with the least payload weight delivered to the Moon. The payloader must perform the following functions:

1) Extend the boom 4 feet to force the payload bucket over the lunar surface until filled. This function is to be performed in 6 seconds and requires 0.5 horsepower to load 1 cubic yard of lunar dust;

2) Rotate the payloader bucket 15 degrees to keep the load retained in the bucket. This function requires 0.4 horsepower and is to be performed in 2 seconds;

3) Elevate the boom through 62 degrees during the time the load is being transported to the dump site. Five horsepower are required to accomplish this task in 6 seconds;

4) Extend the booms to a maximum of 40 feet to reach the dumping height at the top of the caisson. This function requires 2.5 horsepower and is to be accomplished in 15 seconds;

5) Rotate the payload bucket 100 degrees to dump the load. This is to be accomplished in 2 seconds and requires 0.3 horsepower;

6) Return the booms and bucket to their original positions. This can be accomplished in a maximum of 15 seconds with 0.2 horsepower.

About 300 cycles of the above six steps are required to cover one typical lunar shelter. Steps 3 and 4 can be accomplished simultaneously.

These requirements resulted from consideration of the time man can work at such a job and in the relative importance in how fast the job must be accomplished. The selection of 4 hours to cover one shelter of an average height of 40 feet (assuming that the shelter diameter is 2/3 to 3/4 of the vehicle height which includes the vehicle landing propulsion unit) results in the use of a 1-cubic-yard bucket and a boom elevation time of 6 seconds (see Figure 4). Using this boom elevation time and a 4-foot stroke to fill the bucket, the maximum operating power at the load was determined to be 5 horsepower (see Figure 5).

To provide a common basis for evaluation of actuation concepts, each extension boom is defined as a seven-section screw jack — each section extends 68.5 inches. The jack screw rises 1 inch per revolution of the ball screw; the ball screw turns at 2000 rpm. Use of the screw jack is not necessarily the optimum selection for this task. Other possibilities are a scissors jack or a two-section telescoping jack. The screw jack was selected for its compactness. The jack is not considered part of the actuation system because it is common to all concepts evaluated.

The payloader actuation system will be subjected to a temperature environment of 275 to -240°F on the lunar surface; thus, both heating and cooling systems are required.

The payloader attachment is a self-contained unit; therefore, the power system for the unit is supplied for only the payloader task. This power unit is contained within the removable package. Control power for servoactuation and conditioning power for the system and the payloader operator will be supplied by the central motive vehicle.

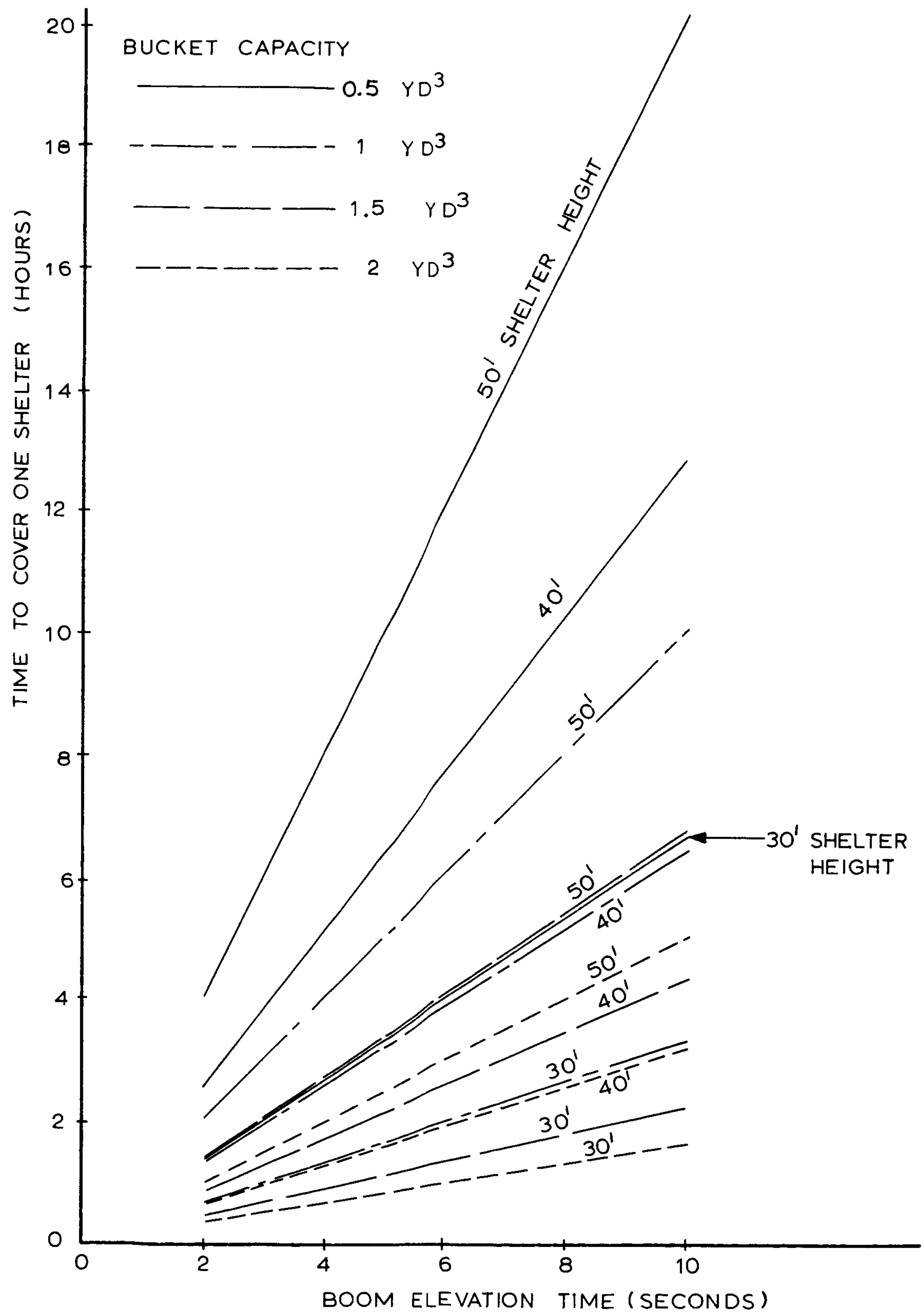


Figure 4: Payloader Design Operation Time

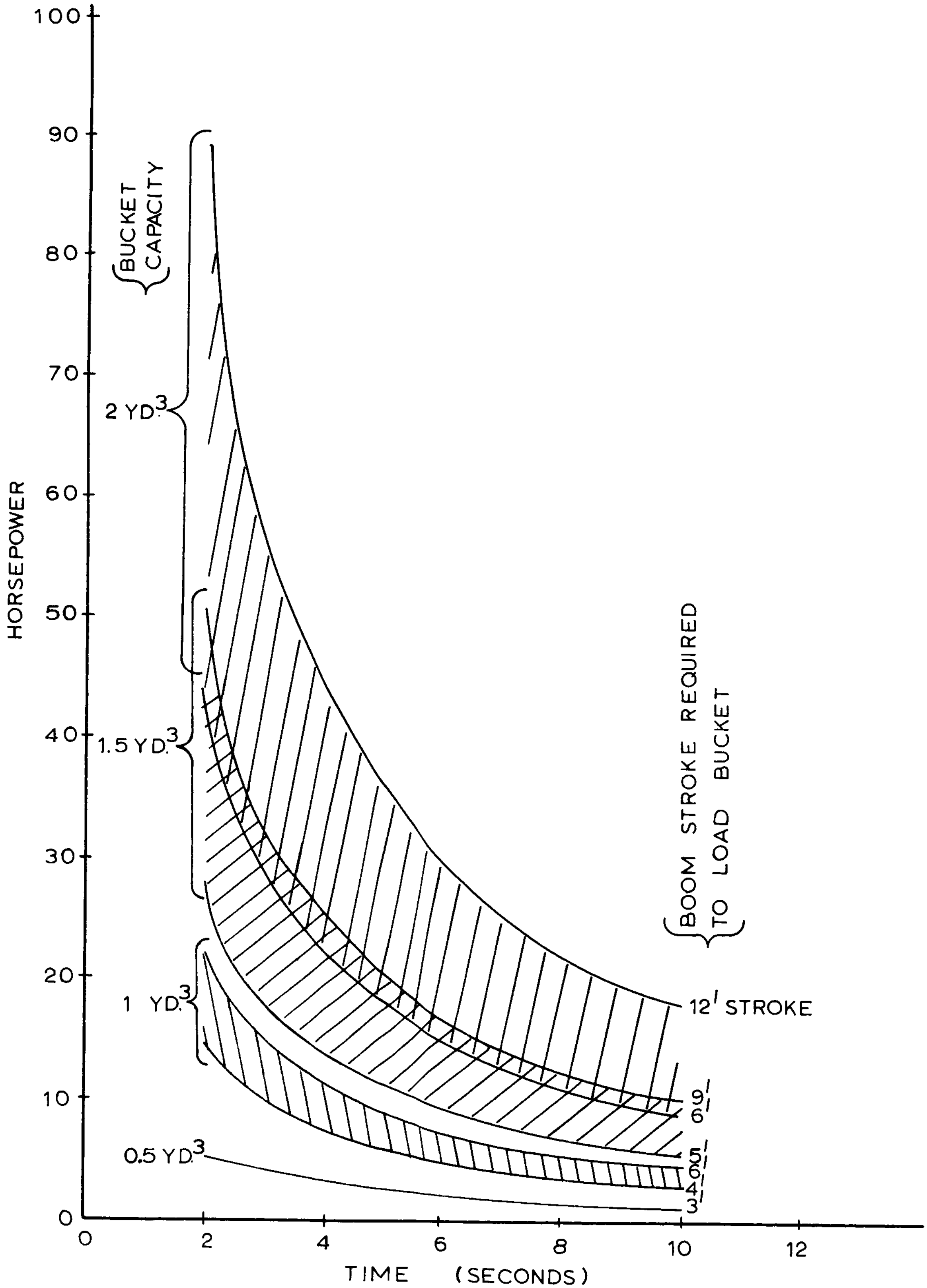


Figure 5: Payloader Design Maximum Power

5.0 ACTUATION SYSTEM DESIGN

After the actuation requirements discussed above were determined, a manufacturer inquiry was prepared to obtain suggested ideas for actuation equipment designs related to the selected tasks. Figure 6 is a list of companies to whom requests were sent and their replies.

Boeing wishes to express particular appreciation to the following suppliers for detailed treatment of design considerations peculiar to the tasks defined in the trade study.

The Bendix Corporation — Products Aerospace Division

Cadillac Gage Company — West Coast Division

General Precision, Incorporated — Kearfott Division

Lear Siegler, Incorporated — Power Equipment Division

Information and data received and used as reference material for the trade study are indicated by an asterisk (*) in Figure 6 and are referenced in the discussions of system designs that follow.

5.1 ELEVON ACTUATION

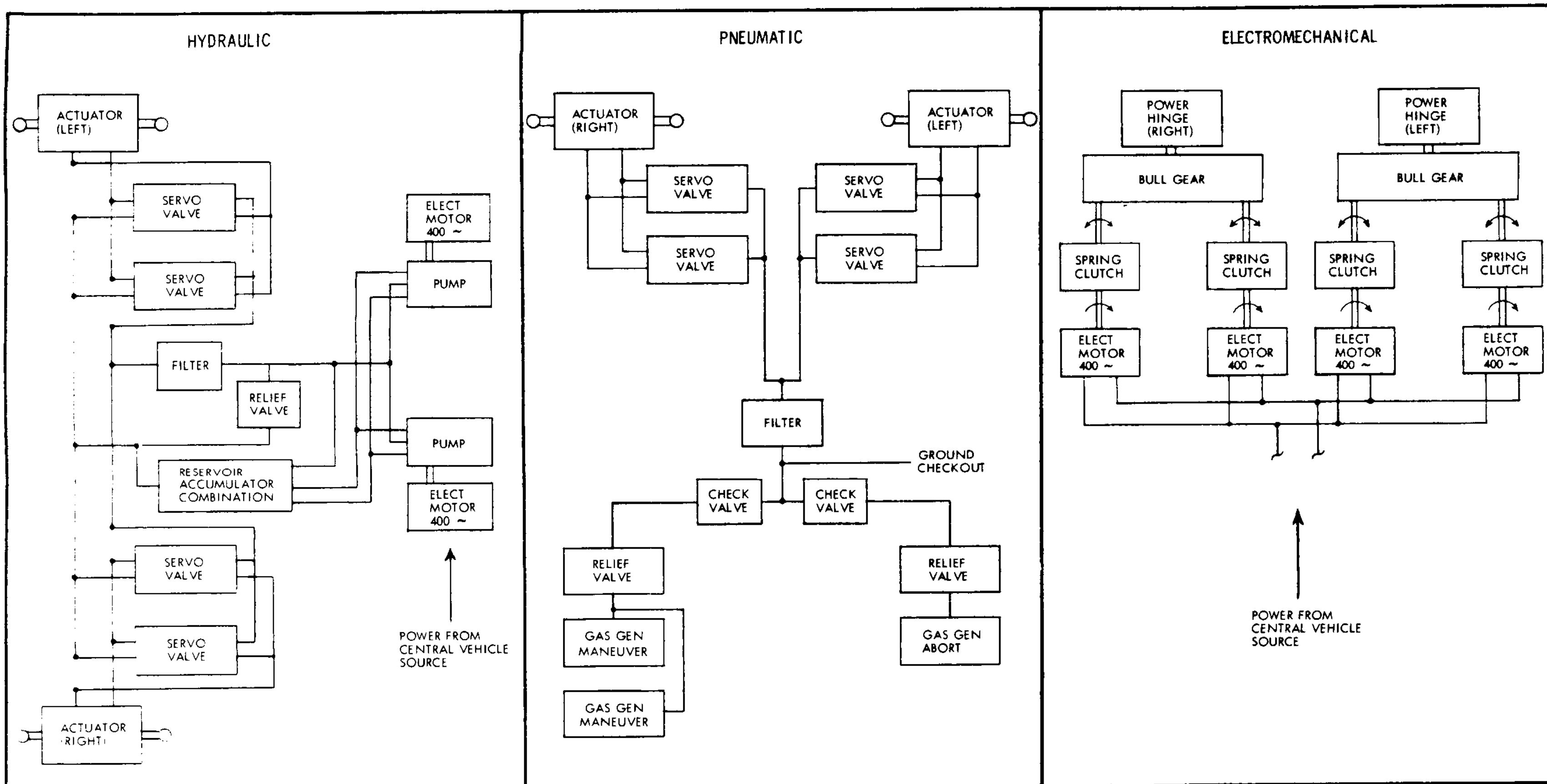
5.1.1 HYDRAULIC CONCEPT

The hydraulic concept for elevon control could incorporate either a linear or rotary drive mechanism. For this study, a linear mechanism was selected as a more direct linkage to the elevon load. Figure 7 shows a schematic block diagram of the system. Each of the two elevons is powered by a double-acting linear actuator that must work against a maximum hinge moment of 144, 000 inch-pounds. This moment occurs during vehicle escape coincident with boost abort. With a minimum lever arm of 5 inches, a 28, 800-pound actuator force is required. A system pressure of 3000 psi was selected because most current systems operate with this pressure. Servovalve design assumes a one-third pressure drop through the valve; this leaves 2000 psi differential pressure available across the actuator piston. This working pressure established the requirement for a piston area of 14.4 square inches. Maximum elevon deflection is ±40 degrees; thus, a 13-inch actuator stroke is needed. The hydraulic flow rate per actuator, based on the deflection rate of 15 degrees per second, is 9.12 gpm. The two elevons must operate simultaneously; therefore, total system flow rate must be 18.24 gpm.

Because the 144, 000-inch-pound load per elevon is experienced only during a short period of maximum dynamic pressure during a boost abort, this requirement can best be satisfied by an accumulator. The accumulator has a working volume of 14 cubic inches, which is sufficient to ensure movement of the actuator pistons to a hard-over position against the maximum dynamic pressure in 0.1 second. The redundant pumps will satisfactorily maintain the hard-over position and replenish the accumulator.

Company	Area of Interest	Response	Applicable Documentation
*American Brake Shoe	Hydraulic/pneumatic	Component brochures	Pump data
*Bendix Corp.	Pneumatic/electromechanical	Pacific Division DC-409; H/P-346 Products Aerospace BPAD-864-15541R, BPAD-864-13A	BPAD-864-15541R (Ref. 16)
Bertea Design	Electromechanical		
*Borg Warner Corp.	Hydraulic/pneumatic	Weston Hydraulic Division Hot-gas brochures Digital actuator Report 756	Pneumatic servovalve data Pneumatic regulator data
*Cadillac Gage	Electromechanical	Proposal 903A	Proposal 903A (Ref. 17)
Chandler-Evans Corp.	Pneumatic	Pneumatic system brochures	
Crane Co.	Hydraulic		
*Curtiss Wright Corp.	Electromechanical	PD-477C, PD-483B, PD-448, EM-0363-026, PD-439, PD-495, PD-468, SD-181	PD-439 (Ref. 12) PD-468 (Ref. 13)
Duff Norton Co.	Mechanical	Component brochures	
Electronic Specialty Co.	Electromechanical		
Garrett Corp.	Hydraulic/pneumatic/electromechanical	AiResearch F-8012	
*General Precision, Inc.	Hydraulic/electromechanical	Kearfott Division A 0914998E	A 0914998E (Ref. 15)
Hydraulic Research	Hydraulic		
*Lear Siegler, Inc.	Pneumatic/electromechanical	Power Equipment Division REA 65-524, REA 64-295, REA 65-520	REA 65-524 (Ref. 14)
Ling-Temco-Vought, Inc.	Electromechanical		
Marquardt Corp.	Pneumatic	Letter	
Minneapolis Honeywell	Hydraulic		
Moog Servocontrols	Hydraulic/pneumatic		
*Parker Hanifan Corp.	Hydraulic/pneumatic	Parker Aircraft component data	H_2 Pneumatic servo EDL 5640022 Accumulator reservoir 5630087
Pneumodynamics Corp.	Hydraulic		
*Rocket Research Corp.	Pneumatic	Letter	Payloader bucket actuation information
Sperry Gyroscope	Hydraulic		
Sterer Engineering	Hydraulic/pneumatic		
Sundstrand Corp.	Hydraulic		
*Talley Industries	Pneumatic	Letter	Solid-propellant gas generator information
TRW Inc.	Hydraulic		
United Aircraft Corp.	Hydraulic		
*Vickers Inc.	Hydraulic/pneumatic	Products Aerospace Divisic component brochures	Relief valve, pump, motor data; ᵔumatic motor data
Walter Kidde & Co.	Pneumatic		

Figure 6: Actuation System Survey



Hydraulic — DESIGN DATA

OPERATING PRESSURE	3000 PSI
MAXIMUM FLOW	18.3 GPM
MINIMUM START-UP TEMP	-60°F
WEIGHT: ACTUATION SYS	210 LBS
CONDITIONING	27 LBS
POWER PENALTY	200 LBS
TOTAL	437 LBS

Pneumatic — DESIGN DATA

OPERATING PRESSURE (MANEUVER)	1000 PSI
(ABORT)	3000 PSI
MAXIMUM FLOW (ABORT)	0.105 LBS/SEC
MINIMUM START-UP TEMP	-60°F
WEIGHT: ACTUATION SYS	104 LBS
NO CONDITIONING	-
POWER SOURCE	100 LBS
TOTAL	204 LBS

Electromechanical — DESIGN DATA

POWER	115 VOLTS 400~
CURRENT	160 AMPS
MINIMUM START-UP TEMP	-60°F
WEIGHT: ACTUATION SYS	210 LBS
CONDITIONING	30 LBS
POWER PENALTY	286 LBS
TOTAL	526 LBS

Figure 7: Re-Entry Vehicle Elevon Actuation Systems

Though hydraulic pumps are a potential point of unreliability, redundant pumps to supply abort flow requirement would be impractical from a power standpoint. The compromise reached for this study was to select a pump size capable of providing two thirds of the abort power requirement. Loss of one pump does not affect maneuver capability, but some loss in high-load performance during abort results. Each of the redundant servovalves has the full-flow capability for the abort condition.

The use of electric motors between the power source and the pumps allows isolation of the pumps, when they are not required, from the power source that operates continuously. It was not considered wise to select a clutching mechanism between the power source and the pumps because of reliability considerations.

The fluid selected for use in the elevon system is MLO 7277, which is satisfactory for a -60 to 400°F environment. Fluid coolers are used in the hydraulic fluid loop to protect the system in the 550°F maximum environment. The environmental control system describing these coolers is discussed in Section 6.1.

A study of the overall power requirements for the shuttle vehicle is reported in Section 7.1. Included is a discussion of the increase in size of the chemical dynamic unit to satisfy hydraulic elevon actuation requirements.

A description of the major components in the system and their approximate sizes follows:

Actuator, double acting, titanium, two required 14.4-sq-in. area by 13-in. stroke Overall dimensions: about 32 in. long by 5 in. dia. with actuator centered.	36 pounds each
Servovalve, flow control, four required 3000 psi supply, 2000 psi diff. Overall dimensions: 1.8 by 1.8 by 3.0 inches	0.75 pound each
Filter, in-line, one required 25-micron nominal Overall dimensions: 3-in. dia by 6-1/16-in. long	4.1 pounds
Reservoir-Accumulator Combination, titanium 950-cu-in. reservoir, 14-cu-in. working accumulator Overall dimensions: 10-in. dia by 20-in. long	19 pounds
Relief Valve, in line, one required 3600 psi relief, 3300 reseat Overall dimensions: 1.4-in. dia by 6.5-in. long	1 pound
Pumps, variable displacement, two required 12 gpm at 8000 rpm, 3000 psi Overall dimensions: 5 by 5 by 9 inches	7 pounds each

Electric Motors, 120-v, 400-cycle, two required 4.5 hp with 300-percent overload for 5 seconds Overall dimensions: 8-in. dia by 12-in. long	19 pounds each
Tubing and Fluid Tubing 1/2- and 3/4-in. sizes, 0.032 wall thickness, corrosion resistant stainless steel Fluid high-temp characteristics with best low temp Suggested fluid: MLO 7277	60 pounds

5.1.2 PNEUMATIC CONCEPT

The pneumatic concept for the elevon control on the manned re-entry shuttle vehicle incorporates a linear mechanism. The system is designed to meet both the abort and flight maneuver requirements with the same servovalves and double acting actuators, but requires separate gas generators. Figure 7 shows a schematic of the system. Detailed requirements are discussed below.

Abort Requirements — Abort requirements determine the actuator size. The pneumatic actuator sizing design selected to satisfy the abort requirements is identical to that of the hydraulic actuator discussed above (i.e., 28,800 pounds of force, 5-inch lever arm, 2000-psi differential pressure, 14.4-square-inch piston). A gas generator capable of producing a system pressure of 3000 psi was selected. The system mass flow rate to supply both left and right actuators is based on the deflection rate of 15 degrees per second and is 0.105 pound per second. A 4-second operating time is required for the abort condition. The equipment supplied to satisfy the abort condition includes redundant servovalves, but does not include redundant abort gas generators. If the abort gas generator fails, the two maneuver generators operating together would be capable of delivering 25 percent of abort power, thereby allowing the abort mission to be completed, but at considerably reduced performance.

Flight Maneuver Requirements — For the flight maneuver, each actuator must operate against a maximum hinge moment of a 50,000-inch-pound load. Based on a 5-inch minimum lever arm, a 10,000-pound actuator force is required. A gas generator capable of producing a system pressure of 1000 psi was selected and, assuming a 300-psi pressure drop through the servovalve, a 700-psi differential pressure is available across the actuator piston. The system mass flow rate for the two actuators is based on the deflection rate of 15 degrees per second (same as for the abort condition) and is 0.04 pound per second. A 10-minute operating time is required for the flight maneuver. Two gas generators, one completely redundant, are installed in the system with redundant servovalves. This redundancy is considered sufficient to man-rate the system for the re-entry function.

Ammonium nitrate is suggested by a rocket propellant manufacturer for the propellant in the gas generators. This propellant is satisfactory for use from -65°F to +160°F. Because the environmental temperature range is anticipated to be -60°F to 550°F, the propellant would be exposed beyond the allowable limits and, therefore, thermal conditioning is required. The high-temperature environment that occurs during re-entry necessitates insulating the propellant grain because the autoignition temperature is between 300 and 350°F.

This insulation will be provided as part of the gas generator. The low-temperature environment, which occurs only during storage at the space station, is not a problem because the solid-propellant rocket gas generators can be removed from the vehicle and stored in a temperature-controlled area. A development program is required to package the system and verify gas system performance.

The descriptions of major components of the system are as follows.

Abort gas generator and relief valve One assembly required, 3000-psi ammonium nitrate propellant	4 pounds
Insulated maneuver gas generator and relief valve Two assemblies required, 1000-psi ammonium nitrate propellant	48 pounds each
Check valves, two required	5 pounds each
Filter, in-line, one required 25-micron nominal	1.5 pounds
Servovalve, flow-control, four required 1/3 pressure drop through valve	1 pound each
Actuator, double-acting, Rene' 41, two required 14.4-sq-in. area by 13-in. stroke	40 pounds each
Tubing and Fittings 3/8- and 1/4-inch line size, 0.032 wall thickness, corrosion resistant stainless steel	8 pounds

5.1.3 ELECTROMECHANICAL CONCEPT

The electromechanical actuation system for shuttle vehicle elevon control consists of one actuator on each of two elevons, powered by alternators driven by the vehicle central power supply. Each actuator consists of redundant, 5-horsepower, a.c. motors and two redundant spring clutch servos that drive a bull gear. Figure 7 shows a schematic of this system. The bull gear for each actuator is attached to the input shaft of a power hinge designed using Reference 12 as a handbook. The output member of the power hinge is directly attached to the elevon structure and is the power transmission element to the elevon. Rotary power transmission was selected for the electromechanical approach as the most direct drive, eliminating conversion to linear drive and reconverting to rotary.

Alternating-current motors were selected because they weigh less than d.c. equipment and because a.c. power is available from the vehicle power supply. The motors were sized to operate briefly at a 50-percent overload under maximum elevon load conditions during boost abort. The vehicle environmental conditioning system, described in Section 6.1, includes provisions for cooling the motors during re-entry.

Spring-clutch servos are similar to those described in Reference 13; each contains two counterrotating spring clutches, either of which can be actuated by a solenoid. The direction of rotation of the servo output shaft is controlled by actuating the appropriate spring clutch. Because the spring clutch, unlike a magnetic particle clutch, does not slip, the elevon exhibits a step-like motion. Spring clutches were chosen for this application because they are lighter than magnetic particle clutches at the power level required for elevon control.

The bull gear connects both spring-clutch servos to the power hinge. The redundant motor and spring-clutch servo are not powered while the primary channel is operating, so the only part of the redundant channel that rotates is the output shaft of the servo, which is connected through the bull gear. The power hinge functions both as a planetary reduction gearbox and as the elevon hinge.

The overall efficiency of the system is 44 percent. The motor efficiency is less than optimum because the motors were sized to provide the abort requirements and operate well below their rated output during re-entry maneuver. Maximum power required from the vehicle supply is during the abort maneuver and is 18.6 kva for both actuators. The weight penalty chargeable to the actuation concept for increasing the size of the vehicle power supply to satisfy actuation requirements is discussed in detail in Section 7.1.

Brief descriptions of the actuation system components are listed below.

Motors, four required 5-horsepower with 50-percent overload for 6 seconds 115/208-v, 400-cycles	23 pounds each
Spring-clutch servos, four required 230 in.-lb at 8000-rpm input shaft speed	10 pounds each
Power hinges, two required 144,000-in.-lb output	30 pounds each
Gears	6 pounds
Electrical Cables	6 pounds

5.1.4 MECHANICAL CONCEPT

The elevon actuation system does not lend itself to being performed by a pure mechanical system that is competitive with the hydraulic, pneumatic, or electromechanical systems discussed. The electromechanical system described above, with the exception of the electrical motors, is pure mechanical. To provide the energy supplied by electric motors in pure mechanical form, such as a flywheel or spring, is not competitive. No further evaluation of the mechanical concept will be discussed.

5.2 CENTRIFUGE ACTUATION

5.2.1 HYDRAULIC CONCEPT

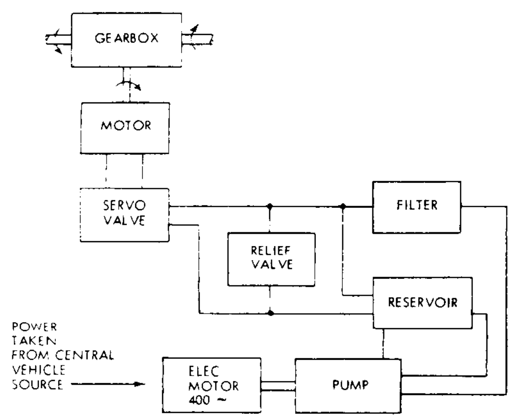
A rotary actuation mechanism is needed to operate a centrifuge. Each of two 30-foot counterrotating wheel assemblies selected for this study can carry four men, which requires 6 horsepower delivered to the wheels through a double-ended gearbox with counterrotating shafts. The gearbox has a 26:1 reduction from the drive motor, which is servo-operated to provide a 0.01- to 6-g acceleration for the crew. Hydraulic supply pressure is assumed to be 3000 psi with a one-third pressure drop through the servovalve. An accumulator is not used because the system is not susceptible to dynamic loading variations. With allowances made for bearing, gearbox, and motor efficiencies, the flow required from the pump is 6.5 gpm. An electrical motor is selected to drive the pump; this provides the least complex power transmission from the central vehicle power supply. The motor must deliver 11.5 horsepower to the pump. Use of the central vehicle power supply is discussed in Section 7.0. Figure 8 shows schematic of the selected concept.

MIL-H-5606 hydraulic fluid is to be used with this system because the equipment compartment is environmentally conditioned. Cooling of the fluid is required because the equipment must operate for extended periods. Cooling fluid for the heat exchanger is supplied from the vehicle environmental control system. These conditioning requirements are discussed further in Section 6.2 of the report.

Descriptions of the major components in the system and their approximate sizes and weights follow.

Gearbox, double-ended, counterrotating shafts, one required 8900-rpm input, 26:1 reduction Overall size: about 4 by 4 by 12 inches	7.5 pounds
Hydraulic motor, variable-displacement, one required 6.5 gpm at 8900 rpm Overall dimensions: 2.5 by 2.5 by 5 inches	4.5 pounds
Servovalve, flow-control, one required 3000-psi supply, 2000-psi differential pressure Overall dimensions: 1.8 by 1.8 by 3.0 inches	0.75 pound
Relief valve, in-line, one required 3600-psi relief, 3300-psi reseat Overall dimensions: 1.4-in. diameter by 6.5-in. long	1.0 pound
Filter, in-line, one required 15-micron nominal Overall dimensions: 3-in. diameter by 6-1/6 in. long	4.1 pounds
Reservoir, bootstrap, one required 44-cu-in. fluid, titanium Overall dimensions: 4-in. diameter by 6.1-in. long	2.0 pounds

HYDRAULIC



DESIGN DATA

OPERATING PRESSURE		3000 PSI
MAXIMUM FLOW		6.5 GPM
MINIMUM START-UP TEMP		CREW ENVR.
WEIGHT:	ACTUATION SYS	50 LBS
	NO CONDITIONING	-
	NEG POWER PENALTY	-
	TOTAL	50 LBS

PNEUMATIC

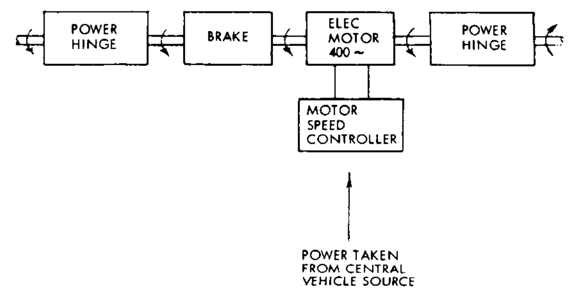
NOT COMPETITIVE

POSSIBLE GAS SOURCES

1. STORED, CLOSED LOOP, RECIRCULATING
2. GENERATED, OPEN LOOP
3. BLEED FROM VEHICLE SUPPLY

ALL POSSIBILITIES WERE REJECTED AS IMPRACTICAL COMPETITION

ELECTROMECHANICAL



DESIGN DATA

POWER		115 VOLTS 400~
CURRENT		49 AMPS
MINIMUM START-UP TEMP		CREW ENVR.
WEIGHT:	ACTUATION SYS	44 LBS
	NO CONDITIONING	-
	NEG POWER PENALTY	-
	TOTAL	44 LBS

Figure 8: Space Station Centrifuge-Actuation Systems

Pump, variable-displacement, one required 6.5 gpm at 8900 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 5 inches	4.5 pounds
Electric motor, 120-v, 400-cycle, one required 11.5-hp output Overall dimensions: 4.5-in. diameter by 5-in. long	18.0 pounds
Tubing and fluid Tubing: 3/8-in. OD by 0.028 wall thickness, corrosion resistant stainless steel Fluid: MIL-H-5606A	6.4 pounds

5.2.2 PNEUMATIC CONCEPT

A gas source is required to operate the centrifuge pneumatically. This source can be made available with a stored gas supply and by operating the system as a recirculating closed loop. The size of the storage tank is impractical when considered in competition to other actuation concepts. The actuation gas could also be supplied from a gas generator using an open loop system. Because the system must operate 45 minutes per day, gas generators would have to be replaced frequently. This method of operation is impractical when compared to other actuation methods. The third possibility of obtaining gas is bleed from a vehicle supply. The space station does not have a known supply of such gas other than for environmental control. Use of this gas for actuation is not considered desirable. Because a practical source of gas for the pneumatic system is not attainable, it is concluded that the pneumatic actuation concept for the centrifuge is not competitive with the other actuaction concepts.

5.2.3 ELECTROMECHANICAL CONCEPT

The electromechanical drive system for the centrifuge performs the same functions as the hydraulic system. The system consists of a 6-horsepower electric motor (with 25-percent short-time overload capability), motor speed controller (frequency changer), and power hinges connected to the centrifuge wheel hubs. Figure 8 shows a schematic of the system. The principle of the motor with controller was suggested in Reference 14 where a regenerative electrical braking system is also suggested. Because regenerative electrical power is of no benefit to the space station, a mechanical brake integral with the motor was selected as a simpler and lighter method of braking. The power hinges (Reference 12) make use of a type of planetary gearing. They make the installation compact and allow the motor to be mounted in line with the centrifuge axis between the wheel hubs. Conditioning to prevent overheating of the electromechanical components will be provided through the space station environmental control loop in a manner similar to the one discussed for the hydraulic concept. Further details are discussed in Section 6.2. Use of actuation power from the vehicle central supply at no penalty to the actuation system is discussed in Section 7.0.

A description of the major components in the system and their approximate size and weights follows.

Motor, one required 6 horsepower with integral friction brake 1200 rpm 115/208-v, 400-cycle	24 pounds
Motor controller, frequency change, one required 5.6-kva maximum output	10 pounds
Power hinges, two required 12,000-inch-pounds maximum torque	5 pounds each

5.2.4 MECHANICAL CONCEPT

A mechanical system design is not feasible for the centrifuge actuation task due to the acceleration and braking requirements.

5.3 THRUST VECTOR CONTROL ACTUATION

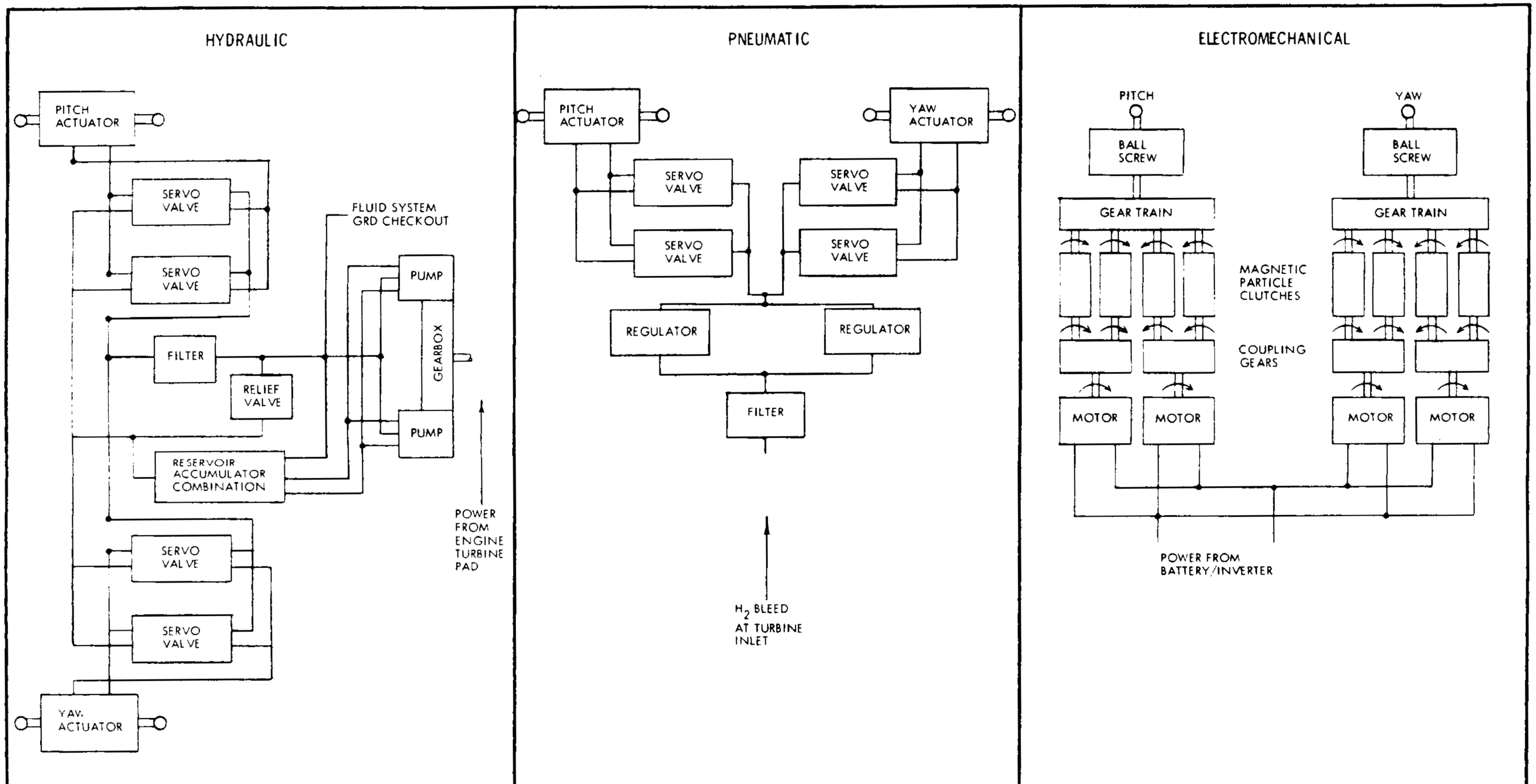
5.3.1 HYDRAULIC CONCEPT

The RL-10 engine gimbal arrangement is designed to be controlled by linear actuators (see Reference 11). Maximum hinge moment (3925 inch-pounds) occurs at maximum engine deflection (8.0 degrees).

A detailed load analysis of this application was performed to establish the hydraulic test system requirements; this analysis is documented in Volume II of this report. The analysis indicates that a 4-inch actuator stroke is preferable with a minimum lever arm of 7 inches. A comparative load analysis of this application is also presented in Reference 15.

With these as requirements, an actuator force of 559 pounds is needed. For a 3000-psi system pressure at the maximum power point for valve design (1/3 pressure drop in the valve), a piston area of 0.279 square inch is required. Maximum actuator flow is 0.44 gpm. The engine vectoring requirement is 20 degrees per second in any direction; therefore, maximum system flow exists when both actuators are used to vector the engine to a 45-degree angle between pitch and yaw. This maximum flow is 1.4 times the flow of one actuator, or 0.62 gpm.

The pump and servovalves are redundant in the system design as shown schematically in Figure 9. Other components are considered incapable of causing a critical failure. Each pump will satisfy the maximum flow requirement of the system with the accumulator added only to smooth out pressure peaks. The gearbox coupling to each pump will incorporate a shear section that will fail if the pump "freezes." This feature provides disconnection of the unusable pump while still allowing the usable pump to operate normally. A fluid system ground checkout is provided because the flight pumps are only operable during RL-10 engine operation. The input gearbox is attached to the RL-10 engine turbine pad which is capable of driving the hydraulic systems without engine modification or power penalty to the hydraulic concept. Section 7.0 contains a further discussion of the power system.



HYDRAULIC — DESIGN DATA

OPERATING PRESSURE	3000 PSI
MAXIMUM FLOW	0.6 GPM
MINIMUM START-UP TEMP	-150°F
WEIGHT: ACTUATION SYS	22 LBS
CONDITIONING	18 LBS
NEG POWER PENALTY	-
TOTAL	40 LBS

PNEUMATIC — DESIGN DATA

OPERATING PRESSURE	600 PSI
MAXIMUM FLOW	.002 LBS/SEC
MINIMUM START-UP TEMP	-240°F
WEIGHT: ACTUATION SYS	17 LBS
NO COND REQD	-
HYDROGEN PENALTY	2 LBS
TOTAL	19 LBS

ELECTROMECHANICAL — DESIGN DATA

POWER	28 VOLTS
CURRENT	26 AMPS
MINIMUM START-UP TEMP	-240°F
WEIGHT: ACTUATION SYS	50 LBS
CONDITIONING	1 LB
BATTERY/INVERTER	9 LBS
TOTAL	60 LBS

Figure 9: Lunar Lander TVC-Actuation Systems

The thermal environment to which the RL-10 gimbal actuation system will be subjected is estimated to be -240 to +100°F. A low-temperature fluid, DuPont E-3, is recommended for this application. The fluid must be warmed to -150°F before startup. This temperature is the estimated minimum operating temperature for the fluid. Section 6.3 presents a description of the environmental control system design.

A description of the major components in the actuation system and their approximate sizes and weights follows.

Actuator, double-acting, titanium, two required 0.28-sq-in. area by 4-in. stroke Overall dimensions: 1.5-in. diameter by 10-in. long when centered	2 pounds each
Servovalve, flow-control, four required 3000-psi supply, 2000-psi differential Overall dimensions: 1.8 by 1.8 by 3.0 inches	0.75 pound each
Filter, in-line, one required 15-micron nominal Overall dimensions: 1.0-in. diameter by 4-in. long	0.3 pound
Reservoir-accumulator combination, titanium, one required 50-cu-in. reservoir, 0.5-cu-in. accumulator working fluid Overall dimensions: 4-in. diameter by 8-in. long	2.4 pounds
Relief valve, in-line, one required 3600-psi relief, 3300-psi reseat Overall dimensions: 1.4-in. diameter by 6.5-in. long	1.0 pound
Pump, variable-displacement, two required 0.65 gpm at 9000 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 4.5 inches	1.8 pounds each
Gearbox, speed-reduction, one required 1.25:1 reduction, 11,300-rpm input speed Overall size: about 1.5 by 6 by 3 inches	2.0 pounds
Tubing and fluid Tubing: 1/4- and 3/8-inch, 0.028 wall thickness, corrosion resistant stainless steel Fluid: DuPont E-3	5.6 pounds

5.3.2 PNEUMATIC CONCEPT

The pneumatic concept for the TVC system on the RL-10 space propulsion engine incorporates a linear mechanism. The two double-acting linear actuators provide the force to gimbal the engine against a maximum hinge moment of 3925 inch-pounds. The linear actuators for the pneumatic TVC system are designed around the same hinge moment, stroke, and lever arm requirements discussed for the hydraulic approach. Figure 9 shows a schematic of the system. Gaseous hydrogen from the vehicle was selected for the pneumatic power source. Two pounds of gas are required for the TVC duty cycle. The gas can be obtained at the inlet to the RL-10 propulsion engine turbine at 350°R and 600 psia. With 600 psi as the system pressure and 200 psi as the pressure drop through the servovalve, a 400-psi differential pressure is available across the actuator piston. This working pressure established the requirement for a piston area of 1.5-square inch. The actuator stroke is 4 inches and the actuator velocity is 6.0 inches per second. The maximum system mass flow rate is 0.002 pound per second to supply both pitch and yaw actuators when gimbaling the RL-10 to a 45-degree angle. Intermittent operation during an 8-minute total operating time is required. Flow and pressure metering devices are redundant in the system to provide man-rated reliability.

Analytical results in Reference 16 advocate an electropneumatic acutator that incorporates a pneumatic motor, transmission, and a ball screw. The actuator weighs about four times more than the proposed piston actuator. The main advantages of the heavier actuator are higher static and dynamic stiffness. Because a detailed stiffness analysis is beyond the scope of this study, and because response requirements were not severe, the evaluation ground rule was adhered to and the approach in Reference 16 was not incorporated.

Descriptions and approximate weights of major components are as follows.

Actuator, double-acting, steel, two required 1.61 sq-in. by 4-in. stroke Approximate size: 2-in. diameter by 10-in. length when centered	3.5 pounds each
Servovalve, flow-control, four required 1/3 pressure drop through valve	1 pound each
Regulator: 0.002 lb/sec, two required	1 pound each
Filter, in-line, one required 25-micron nominal	1 pound each
Tubing and Fittings 1/4- and 3/8-inch sizes, 0.032 wall thickness, corrosion resistant stainless steel	2 pounds

5.3.3 ELECTROMECHANICAL CONCEPT

The electromechanical TVC actuation system consists of two a.c. servoactuators powered by a battery inverter, as shown schematically in Figure 9. Each actuator is designed so that a 0.5-horsepower, 400-cycle motor drives the input members of two counterrotating magnetic particle clutches. Clutch outputs are connected to a ball screw to obtain a linear output motion. The direction of ball screw motion is controlled by energizing one or the other of the counterrotating clutches. The electric motor shaft, rotating at 12,000 rpm, is connected to the clutches through gears that reduce the clutch speed to 3000 rpm. The clutch output shaft is geared to the bull gear at a reduction of 2.08:1. The ball nut is integral with the bull gear, so that it rotates at a maximum rate of 1440 rpm. With a 0.25-inch lead on the ball screw, the maximum linear rate of the ball screw is 6 inches per second. This rate is variable, because the clutches can slip as commanded by the control electronics. Overall efficiency of the actuator is 53 percent.

Each actuator contains a redundant motor and pair of clutches that are not powered while the primary channel is operating. All four clutches are geared to the bull gear; therefore, no redundancy is provided for the bull gear and ball screw.

An alternate design approach is to use a servomotor to drive the ball nut. The motor-clutch design was chosen because it would be difficult to provide redundant servomotors without introducing clutches in the design. Also, the motor-clutch design takes advantage of the kinetic energy of the continuously rotating motor armature, whereas the inertia of the servomotor armature limits the response of the actuator.

Spring clutches could be used in place of the magnetic particle clutches with about the same weight and complexity, but magnetic particle clutches were chosen because they have been tested successfully at the minimum temperature in which the TVC actuators must operate.

The battery-inverter power source was chosen because it is the lightest a.c. power source for the lunar landing vehicle. Alternating current is used because a.c. motors are lighter than d.c. motors and do not require brushes and commutators, which are unreliable and have short life when operating in a vacuum. Reference 17 describes a servoactuator design using d.c. power; it gives a comparison to the selected system.

Except for battery protection thermal environmental control is not required for the electromechanical system if the clutches are disengaged from the load during minimum-temperature startups.

TVC actuation system components are briefly described below:

Motors, four required 0.6 horsepower 115-v, 400-cycle 12,000 rpm	4.5 pounds each
Magnetic particle clutches, eight required 0 to 12 inch-pounds Stationary coil design	1.5 pounds each
Gears, ball screws, and actuators housings	18.0 pounds
Rate generators, four required	0.5 pound each

5.3.4 MECHANICAL CONCEPT

A pure mechanical system is not competitive to satisfy TVC actuation requirements.

5.4 PAYLOADER ACTUATION

5.4.1 HYDRAULIC CONCEPT

The payloader actuation system consists of two separate systems, as shown schematically in Figure 10. The larger system is located within the payloader cab and performs the functions of boom extension and boom elevation. The small system, located on the payloader bucket, is electrically controlled through wiring passing through the payloader booms; it performs the function of tipping the bucket. The actuator drive systems are not speed-modulated; therefore, solenoid on-off control is used in preference to servovalves. The motors and smaller pump are fixed displacement. The larger pump is variable-displacement because it serves two motors that operate simultaneously only part of the time.

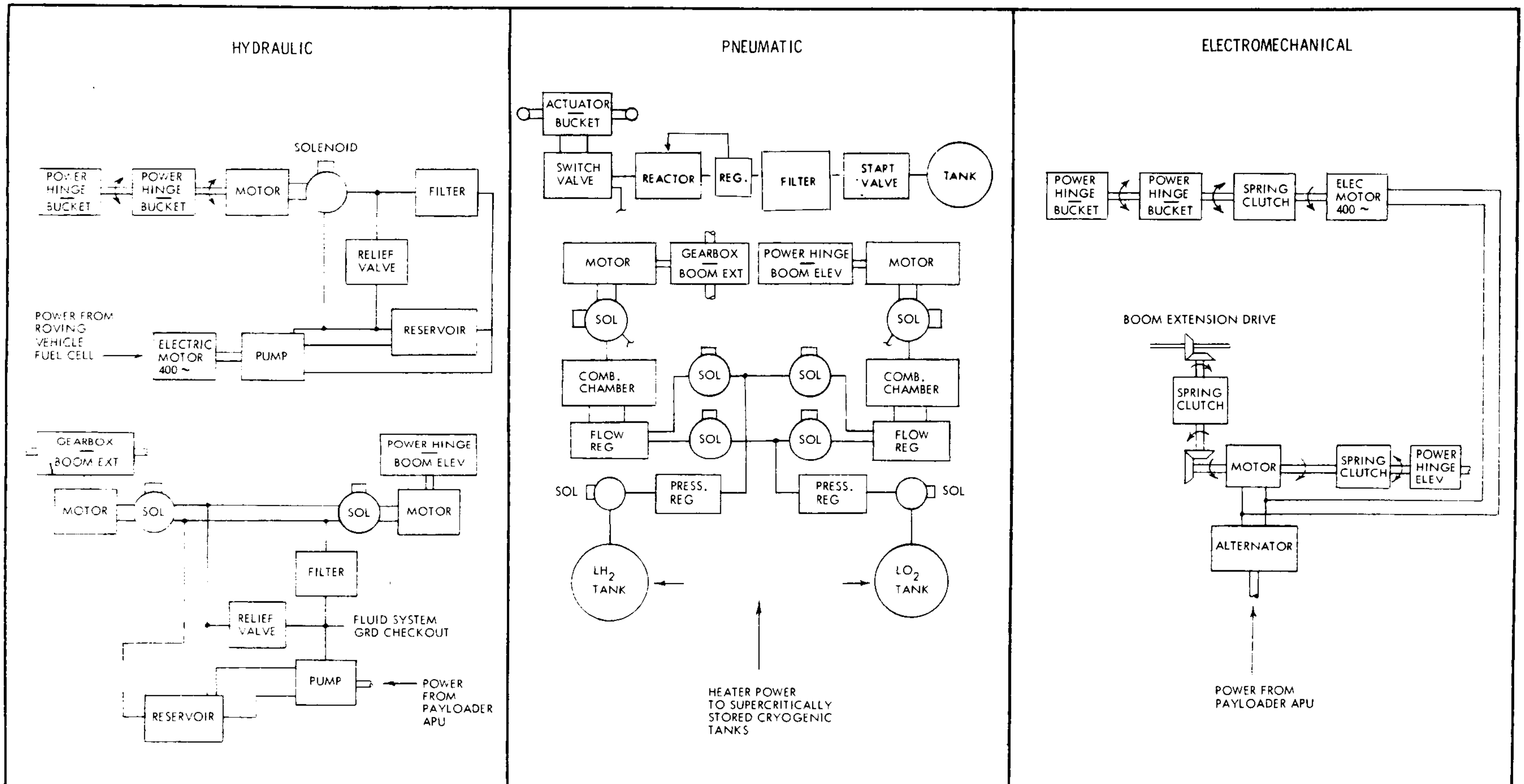
The two boom-extension screw jacks shown in Figure 3 counterrotate; they are driven from a double-ended gearbox with a 6.25:1 speed reduction. The motor must deliver 5.7 horsepower to the gearbox; therefore, for a 3000-psi hydraulic system, 3.25 gpm are required.

The boom-elevation power hinge (see Reference 12) delivers 226,000 inch-pounds of torque to rotate the booms 62 degrees in elevation. The drive motor delivers 7.5 horsepower to the hinge and requires 4.3 gpm at 3000 psi. Maximum delivery from the large pump is 7.6 gpm, which requires 15.6 horsepower from the payloader accessory power unit. This unit is placed within the payloader cab to perform the sole function of supplying power to the larger pump. Requirements for this unit are discussed in Section 7.2.

The payloader bucket is tipped by two power hinges; one hinge is located at the attachment of the bucket to each boom. A single shaft connects these buckets to the drive motor and supplies 0.5 horsepower to the hinges. Pump output flow is 0.32 gpm at 3000 psi. The small hydraulic pump is driven by an electric motor that receives a maximum of 518 watts of power from the motive vehicle fuel cell. This is a practical power source because the bucket is tipped when the vehicle is not moving and wheel power is not demanded from the fuel cell.

Because the payloader thermal environment will vary from -240 to +275°F, actuation equipment must be thermally conditioned for both heating and cooling. DuPont E-3 fluid is selected for the system because it is usable within a larger portion of the temperature range specified than other candidate fluids. Heating is needed to precondition the equipment to -150°F, which is the expected minimum operating temperature of the fluid. Cooling is necessary to keep the fluid below +150°F; above this temperature, the viscosity makes the fluid difficult to pump. Details of the conditioning system are discussed in Section 6.4.

A description of the major components of the actuation system and their approximate sizes and weights follows.



HYDRAULIC — DESIGN DATA

OPERATING PRESSURE		3000 PSI
MAXIMUM FLOW	(EXT/ELEV)	7.5 GPM
	(BUCKET)	0.3 GPM
MINIMUM START-UP TEMP		-150°F
WEIGHT:	ACTUATION SYS	97 LBS
	CONDITIONING	
	POWER PENALTY	258 LBS
	TOTAL	383 LBS

PNEUMATIC — DESIGN DATA

OPERATING PRESSURE		500 PSI @ BUCKET
		300 PSI @ EXT-ELEV
MAXIMUM FLOW	(EXT/ELEV)	0.0043 LB/SEC
	(BUCKET)	0.001 LB/SEC
MINIMUM STARTUP TEMP.		-150°F
WEIGHT:	ACTUATION SYS.	277 LBS
	CONDITIONING	3 LBS
	NEG. POWER PENALTY	
	TOTAL	280 LBS

ELECTROMECHANICAL — DESIGN DATA

POWER		115 VOLTS 400 ~
CURRENT		125 AMPS
MINIMUM START-UP TEMP		-150°F
WEIGHT:	ACTUATION SYS	169 LBS
	CONDITIONING	44 LBS
	POWER PENALTY	269 LBS
	TOTAL	482 LBS

Figure 10: Lunar Rover Payloader-Actuation Systems

Boom extension gearbox, speed-reduction, one required 6.25:1 reduction, 12,500 rpm input Overall dimensions: 6 by 6 by 12 inches approximately	10.0 pounds
Boom extension motor, fixed-displacement, one required 3.26 gpm at 12,500 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 4.5 inches	1.7 pounds
Boom extension solenoid, four-way, one required 3.26 gpm Overall dimensions: 1.4 by 5 by 5 inches	2.0 pounds
Boom elevation power hinge, one required 226,000 in.-lb torque Overall dimensions: 7-in. diameter by 6-in. length	39.0 pounds
Boom elevation hydraulic motor, fixed-displacement, one required 4.3 gpm at 12,500 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 4.5 inches	1.7 pounds
Boom elevation solenoid valve, four-way, one required 4.3 gpm Overall dimensions: 1.4 by 5 by 5 inches	2.0 pounds
Filter, in-line, one required 15-micron nominal Overall dimensions: 1.75-in. diameter by 6.0-in. length	3.0 pounds
Relief valve, in-line, two required 3600-psi relief, 3300-psi reseat Overall dimensions: 1.4-in. diameter by 6.5-in. long	1.0 pound each
Reservoir, bootstrap, titanium, one required 109 cubic inches Overall dimensions: 5-in. diameter by 5.5-in. long	2.1 pounds
Pump, variable-displacement, one required 7.6 gpm at 5200 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 5 inches	4.5 pounds
Bucket power hinge, two required 9000-in.-lb torque Overall dimensions: 3-in. diameter by 1.8-in. wide	2.2 pounds each
Bucket hydraulic motor, fixed-displacement, one required 0.32 gpm at 9300 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 4.5 inches	1.3 pounds
Bucket solenoid, four-way, one required 0.32 gpm Overall dimensions: 1.4 by 5 by 5 inches	2.0 pounds

Filter, in-line, one required 25-micron nominal Overall dimensions: 0.68-in. diameter by 1.5-in. long	0.1 pound
Reservoir, bootstrap, titanium, one required 30 cubic inches Overall dimensions: 3-in. diameter by 4.5-in. long	0.9 pound
Pump, fixed-displacement, one required 0.32 gpm at 10,000 rpm, 3000 psi Overall dimensions: 2.5 by 2.5 by 4.5 inches	1.3 pounds
Electric motor, 115-v, 400 cycle, one required 0.6 hp at 10,000 rpm Overall dimensions: 3.62-in. diameter by 4.25-in. long	4.3 pounds
Tubing and Fluid Tubing: 1/4- and 3/8-inch, 0.028 wall thickness, corrosion resistant stainless steel Fluid: DuPont D-3	14.3 pounds

5.4.2 PNEUMATIC CONCEPT

The pneumatic concept for the payloader boom extension, boom elevation, and bucket rotation systems is shown schematically in Figure 10. An oxygen and hydrogen system was selected for the boom extension and boom elevation system and a hydrazine system was selected for the bucket rotation system. The consideration of storage of a pneumatic system gas source to be used continuously over a 4-hour period on the lunar surface is the primary concern in selecting a pneumatic approach. Storage for operation over this period of time will allow completion of the caisson coverage for one lunar shelter. A new gas supply is required for each additional shelter to be covered.

The hydrogen and oxygen gas supplies are stored at about 400 psi; oxygen is stored in the subcritical state and the hydrogen in the supercritical state. When heated, these fluids become gaseous and pass through control and flow regulator valves into the gas combustion chamber. An oxidizer-to-fuel ratio of 0.64:1 was selected; gas combustion chamber temperature should be about 1000°F. For a 4-hour operating period and a maximum flow rate of 0.0043 pound per second, the useful weight of oxygen and hydrogen to be consumed is 24.2 and 37.8 pounds, respectively. Solenoid valves are used to provide on-off control of the gases to the combustion chamber. Other solenoids are used to control the direction of rotation of the pneumatic motors.

Based on information received from Vickers, Inc., existing pneumatic motors can be redesigned to operate at 300 psi. A thermal conditioning system is required because the motors cannot endure long operating periods at 1000°F. The extension gearbox and the elevation power hinge perform the same functions as described for the hydraulic discussion.

Thermal conditioning of the pneumatic motors is required and is provided by using a heat transport loop. The heat is transferred to the cooling fluid, which in turn is used to warm cryogenic hydrogen. Further explanation of the environmental control design is provided in Section 6.4.

The mechanism designed to tip the bucket (Figure 10) is an open-loop hydrazine system completely installed on the payloader bucket. The hydrazine tank with integral nitrogen pressurization has sufficient ullage to maintain fuel supply pressure for 4 hours of use. Nitrogen prepressure is 1600 psia. Tank replacement or refueling will be required for each additional shelter coverage task. The Shell 405 catalyst in the reactor chamber provides restart capability for intermittent use of the system. Hydrazine dissociation over the catalyst bed provides clean gas at 1800°F and 500 psi for flow rates up to 3 scfm, which is needed to dump the bucket in 2 seconds. The system design is based on research being conducted by a reaction control system supplier. The actuator design incorporates a center-locking function with a double-acting piston that has unequal plus and minus strokes and unequal areas in the plus and minus directions. The switching valve has no modulating capability but only ports gas to provide direction or centering control. The use of the hydrazine system requires thermal environmental control because even with additives, hydrazine will freeze at about -60°F. Provisions for this control are discussed in Section 6.4.

The description of the major components and their approximate weights are as follows.

LO_2	24.2 pounds
LO_2 tankage, 12-in.-dia sphere, one required	8.2 pounds
LH_2	37.8 pounds
LH_2 tankage, 36-in.-dia sphere, one required	72.2 pounds
Pressure regulator, two required	1.0 pound each
Flow control valve, two required	3.0 pounds each
Tachometer assembly and speed control, two required	2.0 pounds each
Solenoid valves, four-way, two required	2.0 pounds each
Gas generator, two required	7.5 pounds each
Shutoff valves, six required	0.5 pound each
Boom extension motor, one required	15.0 pounds
Boom extension gearbox, speed reduction, one required	10.0 pounds
Boom elevation motor, one required	22.0 pounds
Boom elevation power hinge, one required 226,000 in.-lb torque Overall dimensions: 7-in.-dia by 6-in. long	39.0 pounds

Tubing and fittings 1/4-, 3/8-, and 1/2-inch line sizes, 0.028 wall thickness, corrosion resistant stainless steel	4.0 pounds
Bucket tipping actuator, one required	3.0 pounds
Switching valve, one required	2.0 pounds
Hydrazine feed and gas generation system, one unit required	4.0 pounds

5.4.3 ELECTROMECHANICAL CONCEPT

Payloader actuation tasks are performed by two systems: a system to evaluate and extend the boom and a system to rotate the bucket. These systems are shown schematically in Figure 10. The boom actuation system is powered by a 13-horsepower, 8000-rpm motor that runs continuously and provides power for both the extension and elevation functions. A single a.c. motor is used to minimize the size and complexity of the system. Spring clutches engage the motor shaft with the boom extension ball screw and the power hinge that elevates the boom. The spring clutches have a bidirectional output controlled by an electrical command signal. The clutches act as friction brakes to hold the boom in the desired position.

The bucket actuator consists of a 0.5-horsepower a.c. motor, a spring clutch, and power hinges on the bucket to provide speed reduction. Spring clutches are used in both systems to provide for reversal of shaft rotation and load braking. Because a single motor is used for both boom actuation functions, the clutches are required so that these functions can be performed independently. Clutches are necessary also to uncouple the motors from their loads when the motors are started at minimum temperature.

No thermal conditioning of the actuation equipment is required at low temperatures, but the motors and clutches are cooled to limit the maximum temperature to +400°F.

The maximum power required from a power source designed specially for the payloader task is 22 horsepower. This power supply is considered part of the actuation system. Its design requirements are discussed in Section 7.2.

Components for the payloader actuation system are listed below.

Alternator, one required 115/208-v, 400-cycle 15 kva output (maximum)	29 pounds
Motor, a.c., one required 115/208-v, 400-cycle 8000 rpm 13 horsepower	45 pounds

Motor, a.c., one required 115/208-v, 400-cycle 8000 rpm 0.5 horsepower	4 pounds
Bucket power hinges, two required 9000 in.-lb torque	5 pounds
Boom elevation spring clutch, one required 7.5-horsepower capacity	15 pounds
Boom power hinge, one required 226,000 in.-lb torque	39 pounds
Boom extension spring clutch, one required 6-horsepower capacity	12 pounds
Bucket drive spring clutch, one required 0.5 horsepower	5 pounds
Gears and housings	10 pounds

5.4.4 MECHANICAL CONCEPT

A mechanical system that could perform a portion of the payloader task is limited to extension and elevation functions. This system would be that portion of the electromechanical system shown in Figure 10 without the alternator and electric motor. Shaft power would be supplied to the system from an accessory power source.

A direct mechanical system to tip the bucket on the payloader is not practical. Because an entire payloader system by mechanical operation is not possible, the system was not considered further in the trade study.

6.0 CONDITIONING SYSTEM DESIGN

Environmental conditioning requirements must be included in the evaluation of actuation concepts for space environments. The evaluation of all environments that would be encountered in an actual application cannot be accomplished within the limited scope of this study. However, because the primary emphasis of this study is on operational performance in severe temperatures, the evaluation of thermal protection provisions is included. Some consideration is also given to the protection provisions advisable for hard-vacuum operation.

Individual thermal conditioning system designs developed for this study are based upon the following ground rules.

1) Insulation is a multilayer type. Density of the insulation is 5 pounds per cubic foot with an effective thermal conductivity of 4.0×10^{-5} Btu/hr-ft^2-°F/ft. The 5 pounds per cubic foot includes the installation weight for attaching the insulation to the components. The insulation weight may be increased by as much as 50 percent for some applications, such as hydraulic lines. This factor is used both to account for increased insulation thickness required to compensate for attachment heat shorts and increased attachment weight for applying the insulation.
2) Optimum weights are selected where the trade between weight and cost or development time exists.
3) Environments:
    a) For shuttle vehicle elevon control, environment temperature is -60 to +550°F;
    b) For space station centrifuge drive, environment temperature is the same as the crew conditioning temperatures;
    c) For RL-10 engine thrust vector control, environment temperature is -240 to +100°F;
    d) For the payloader, environment temperature is -240 to +275°F.
4) Although detailed reliability studies are not made, some components are redundant on the elevon and thrust vector control actuation tasks since both must be man-rated. The centrifuge and payloader can be maintained and redundancy is not required.

Individual sections that follow provide detailed discussions of the environmental conditioning required for the four tasks studied. Thermal protection requirements varied considerably and were dependent on the task, as shown below.

1) Shuttle vehicle elevon control — Cooling is the only thermal conditioning required.
2) Space station centrifuge drive — Cooling is the only thermal conditioning required.
3) RL-10 engine thrust vector control — Heating is required for some of the actuation methods.

4) Payloader — Both heating and cooling are required for the hydraulic and electromechanical actuation methods. The pneumatic actuation method requires only heating.

Design criteria for equipment to operate in a hard-vacuum environment were considered in the following order.

1) Use equipment designed to endure long term exposure to hard-vacuum.
2) Place actuation equipment in pressurized vehicle compartments that receive the pressurization from the crew conditioning systems.
3) Protect individual pieces of equipment by installation in pressurized canisters.

6.1 ELEVON CONTROL SYSTEM

6.1.1 PROTECTION FOR HYDRAULIC CONCEPT

The elevon system on the shuttle vehicle has an operating environment of -60 to +550°F. The minimum temperature was selected after reviewing the orbital spacecraft skin temperatures associated with various values of solar absorptivity and infrared emissivity. It was assumed that a high emissivity would be required for the re-entry portion of the mission and, by selecting a suitable solar absorptivity, the minimum temperature would be above -100°F. With this transient minimum temperature of -100°F, the hydraulic fluid would not cool below -60°F. The minimum temperature of -60°F allows a Class III (400°F) fluid to be used without any preheating of the equipment. The only thermal protection will be the cooling system, as shown schematically in Figure 11. Redundant heat exchangers satisfy the man-rating reliability requirement.

Hydrogen is used as the heat sink because hydrogen is also used as fuel for the power subsystem. The hydrogen for both power and cooling are stored in the same tank. Only a few pounds of hydrogen are required; therefore, no attempt will be made to use the warmed hydrogen for the power subsystem. Weight penalties for hydrogen and tankage are the same as used for the power subsystem discussed in Section 7.1.

The hydraulic system is not used prior to the last 10 minutes of flight. A hydraulic peak pump load of 23 horsepower for 4 seconds (boost abort escape) and also an average load of 4.5 horsepower for the entire 10 minutes of maneuvering during re-entry must be considered for design of the cooling subsystem. The thermal capacitance of the hydraulic system is sufficient to allow the 4-second abort peak load without additional cooling provisions. The system will also receive aerodynamic heating which has been estimated as approximately 8500 Btu/hr, resulting in a 20,000-Btu/hr total load.

A 350°F hydraulic fluid temperature is considered the maximum allowable at the entrance to the heat exchanger. Average temperature is 325°F. With an assumed hydraulic heat transfer film coefficient of 25 Btu/hr-ft^2-°F and an average heat exchanger plate temperature of 50°F, the heat-transfer area for the heat exchanger can be calculated as follows.

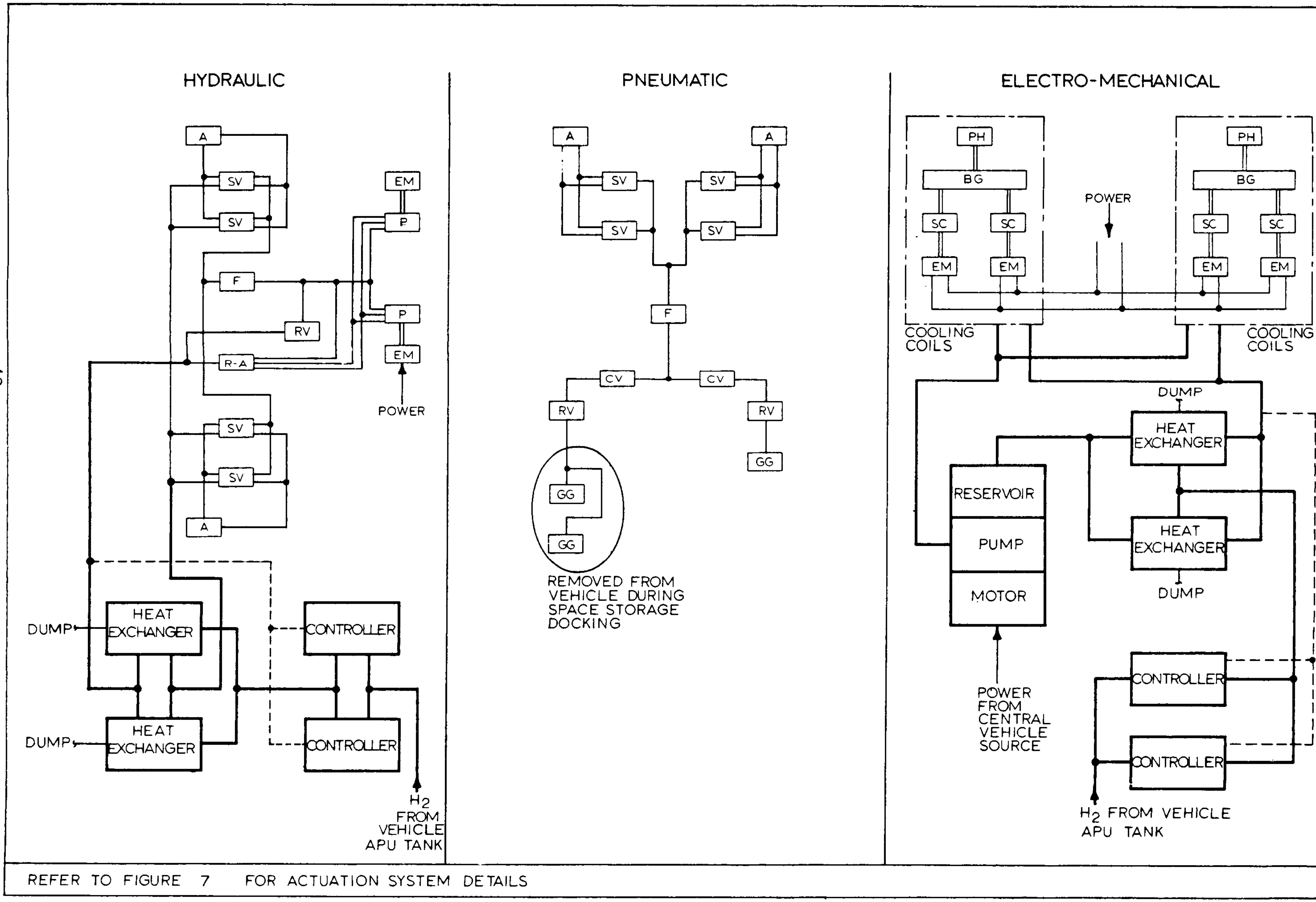


Figure 11: Thermal Protection For Re-Entry Vehicle Elevon Actuation Systems

$$Q = hA\Delta T \tag{1}$$

where

Q = Heat transferred, Btu/hr

h = Heat transfer film coefficient, Btu/hr-ft^2-°F

A = Heat transfer area, ft^2

ΔT = Temperature difference, °F

$$A = \frac{Q}{h\,\Delta T} = \frac{20,000}{(25)(325-50)} = 2.9 \text{ square feet}$$

A small shell and tube heat exchanger will be suitable for this application. The exchanger with 2.9 square feet of heat-exchange area will weigh 6.5 pounds and contain 3.0 pounds of fluid.

If the hydrogen is heated to about 100°F, the average utilization will be about 1670 Btu/lb. The cooling load is 1910 Btu; therefore, 1.1 pounds of usable hydrogen are required. A total hydrogen penalty of 2.0 pounds will be used to provide the usable quantity of 1.1 pounds.

The cooling load is so small and the time is so short that insulation on the lines could not be justified on the basis of reducing the cooling load. Nevertheless some insulation should be put on the lines to prevent local hot spots. With an assumed insulation thickness of 0.25 inch, a total of 1.0 pound is required. A 50-percent factor will be assumed for installing the insulation on the tubing, resulting in a 1.5-pound installed weight penalty.

Following is a description of the heat exchanger and controller. A parallel-cross-flow (shell and tube) heat exchanger and mechanical thermal control valve are required.

Maximum heat exchange: 20,000 Btu/hr

Minimum heat exchange: 2000 Btu/hr

At maximum heat exchange, hydraulic fluid conditions are:

Inlet temperature: 350°F

Outlet temperature: 300°F

Flow rate: 23 pounds per minute

ΔP (allowable): 20 psi

For the above conditions, the hydrogen heat transfer fluid conditions are:

Inlet temperature: -340°F

Outlet temperature: 100°F

Flow rate: 12 pounds per hour

ΔP (allowable): 50 psi

At minimum load, a controller maintains the hydraulic temperature at no less than 0°F by metering hydrogen through the exchanger. A mechanical thermal controller is used. A weight summary for the thermal conditioning system is as follows.

	Pounds
Heat exchanger, two required	13.0 total
Temperature controllers, two required	2.0 total
Fluid, contained in above exchangers	6.0
Insulation	1.5
Hydrogen	2.0
Tank weight penalty	2.5
Total	27.0

The results of the extended-duration vacuum storage test reported in Volume II indicate that protection of the hydraulic system from the hard-vacuum environment is not required.

6.1.2 PROTECTION FOR PNEUMATIC CONCEPT

The pneumatic actuation system for the elevon control on the shuttle vehicle uses solid-propellant gas generators. The propellant must be stored at temperatures that will prevent cracking or severe changes in burning rate. During shuttle vehicle flight, the crew compartment environment and insulation supplied with each generator will protect the generators. When the shuttle vehicle is docked at a space station, other protective measures are required. The simplest and least costly provision is to remove the gas generators that are to be used during the re-entry portion of the mission from the shuttle vehicle and store them in the thermally controlled space station (see Figure 11). By using this procedure, no special conditioning equipment for the pneumatic elevon actuation system is required on board the shuttle vehicle.

Protection considerations for operation of the pneumatic system in a hard-vacuum environment are considered negligible. Outgassing of seals that were a consideration with the hydraulic system will not occur because high-temperature pneumatic seals will be metallic.

6.1.3 PROTECTION FOR ELECTROMECHANICAL CONCEPT

The elevon actuation system must operate within the environmental range of -60 to +550°F. The electromechanical actuators are capable of starting in the minimum temperature environment and, therefore, only a cooling system is required for the maximum temperatures. A schematic of the system is shown in Figure 11. A redundant design is used for this system for the man-rating requirement. The design considers a peak actuation load of 7.2 kilowatts; however, this peak is of such a short duration (during abort) that the thermal capacitance is sufficient and cooling is not designed for this condition. The electromechanical system has a cooling load of 2460 Btu/hr for both actuators. The actuators will also receive some aerodynamic heating, which has been estimated as about 2500 Btu/hr, resulting in a total load of 5000 Btu/hr.

The heat generated at the actuators is transferred to the hydrogen heat sink by using a heat transport loop. The relatively small cooling load will require only a few pounds of hydrogen, therefore, this hydrogen will not be returned to the power subsystem. The hydrogen for this cooling will be stored in the power subsystem main tank, and, therefore, tank penalties will correspond with those used by the power subsystem described in Section 7.1.

With a heat transport fluid inlet temperature to the heat exchanger of 350°F (based on a maximum actuator temperature of 450°F), the flow rate was sized to provide a 300°F fluid outlet temperature from the exchanger. The heat transport fluid is the same as the fluid in the hydraulic system in Section 6.1.1.

If the maximum pressure drop in the cooling circuit is 50 psi and the overall efficiency of the pump and motor is 20 percent, the heat transport pump power is 40 watts for a flow rate of 5.5 pounds per minute.

With an assumed heat transfer film coefficient of 25 Btu hr-ft^2-°F and with 50°F as an average heat exchanger plate temperature, the heat transfer area can be calculated from Equation (1).

$$A = \frac{Q}{h\Delta T} = \frac{5000}{(25)\,(325-50)} = 0.7 \text{ square foot}$$

The total cooling load is 410 Btu's during the 10-minute re-entry. Hydrogen is metered through the exchanger to maintain the heat transport fluid outlet temperature at about 300°F. Assuming the hydrogen is heated to a minimum of 100°F the hydrogen utilization will be about 1670 Btu/lb for the 410 Btu cooling load, 0.25 pound of hydrogen is required. A stored weight of 0.3 pound of hydrogen is used with a 0.4 pound tankage penalty.

The heat transport pumps and reservoir will be similar to equipment presently available for NASA spacecraft programs. Each of the two pumps has a weight (including reservoir) of 8 pounds. Ten feet of fluid lines are required between the exchangers and each actuator plus 5 feet of fluid line at the actuators. The lines will be 0.375 inch in diameter. These lines do not require insulation.

Following is a description of the major conditioning system components.

1) Exchanger and controller

   A parallel-cross-flow (shell and tube) heat exchanger and mechanical thermal control valve are required.

   Maximum Heat exchange: 5000 Btu/hr

   Minimum heat exchange: 200 Btu/hr

At maximum heat exchange the heat transport fluid conditions are:

Inlet temperature: 350°F

Outlet temperature: 300°F

Flow rate: 5.5 pounds per minute

ΔP (allowable): 20 psi

For the above conditions, the hydrogen fluid conditions are:

Inlet temperature: -340°F (maximum)

Outlet temperature: 100°F

Flow rate: 1.5 pounds per hour

ΔP (allowable): 50 psi

At minimum load, the controller will maintain the heat transport fluid between 200 and 300°F leaving the exchanger. This will be done by metering the hydrogen through the exchanger.

2) Heat transport pump and reservoir

An electrically operated gear pump with integral reservoir will be provided. The pump will provide 5.5 pounds per minute of a fluid with a maximum normal pressure drop of 50 psi. The electric motor will be sized to provide a 100-psi head for low-temperature starting. The integral reservoir will contain 0.3 pound of fluid.

The weight summary for the thermal conditioning system is as follows.

	Pounds
Heat exchangers, two required	3.5
Pumps and motors, two required	16.0
Temperature controllers, two required	2.0
Hydrogen	0.3
Tank weight for hydrogen storage	0.4
Tubing	2.0
Heat transport fluid	5.6
Total	29.8

The protection of electromechanical equipment for operation in a hard-vacuum environment appears to be no more serious than the protection of the hydraulic system. No protection will be provided for the electromechanical system.

6.2 CENTRIFUGE CONTROL SYSTEM

Actuation systems designed to operate the centrifuge in the large manned space station will be installed in an environmentally controlled compartment. No preheating for either the hydraulic or electromechanical design is required. Cooling of the hydraulic fluid and motor of the electromechanical system is required because the equipment must operate continuously for 45 minutes per day. The cooling is provided by the liquid cooling loop of the space station, which also provides thermal conditioning of experiment subsystems and other high-power equipment (see Figure 12). It is assumed that the cooling required during centrifuge operation is no more severe than the cooling required for equipment that would normally be operated by the men receiving physiological treatment on the centrifuge. Therefore, no additional thermal conditioning provisions other than those planned for the space station will be necessary. Because all of the centrifuge actuation equipment is located within the vehicle, no vacuum protection is required.

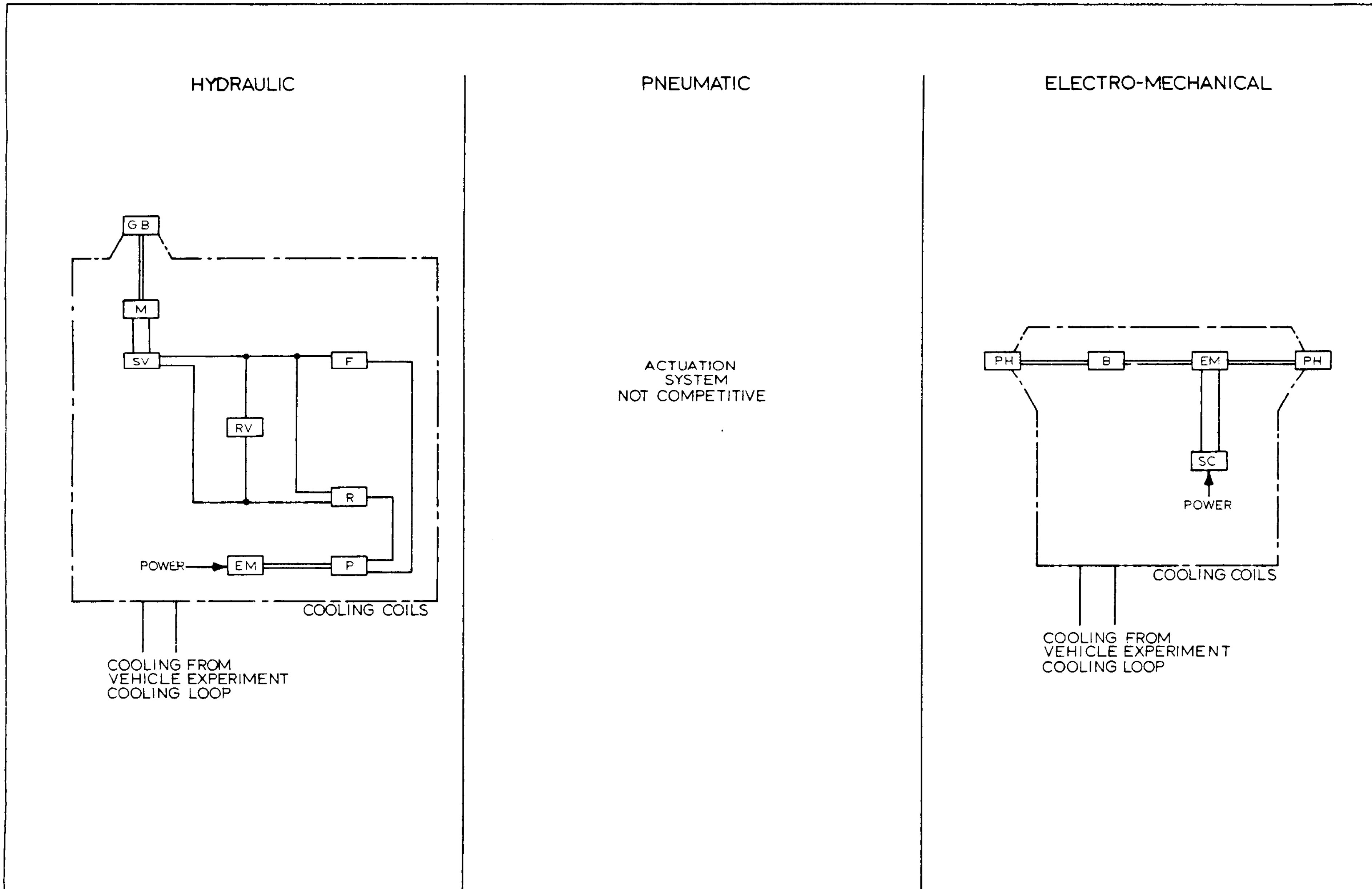
6.3 THRUST VECTOR CONTROL SYSTEM

6.3.1 PROTECTION FOR HYDRAULIC CONCEPT

The thermal protection subsystem for the RL-10 actuation task considers the effect of system operation for both the minimum temperature of -240°F and the maximum temperature of 100°F. This range is compared to hydraulic fluid operational range of -150°F to approximately 150°F based on the fluid viscosity evaluations described in Volume II for the DuPont E-3 fluid. The minimum environmental temperature is the critical concern because it is assumed that the skin of the cryogenic propulsion stage will be coated to maintain minimum temperature to reduce the heat leak into the cryogenic propellant tanks. The maximum environmental temperature does not establish a cooling requirement because the heat developed in the system during a typical 100-second operational sequence will not raise the temperature above the operational limit of the fluid.

Two methods were considered to condition the system when it was exposed to the minimum temperature. These are: (1) maintain an operational temperature continuously through the mission using a battery or isotope energy source, or (2) preheat the actuation equipment before periods of use. The results of a weight comparison indicated 18 pounds for continuous conditioning using plutonium 238 as the isotope, more than 60 pounds for continuous conditioning using a battery, and 19 pounds using a battery for preheating. Preheating was selected for this study. The plutonium isotope system weight is nearly the same weight as the battery preheat system, however, more development is required for the isotope system. The isotope system weight is based on three sources, one for each actuator and one for the power package. If more than two preheats were required, the isotope system would weigh less than the battery preheat system. Figure 13 shows the schematic of the system.

The thrust vector control system is used intermittently during midcourse and lunar landing maneuvers. Equipment operation during these maneuvers generates adequate



REFER TO FIGURE 8 FOR ACTUATION SYSTEM DETAILS.

Figure 12: Thermal Protection For Space Station Centrifuge Actuation Systems

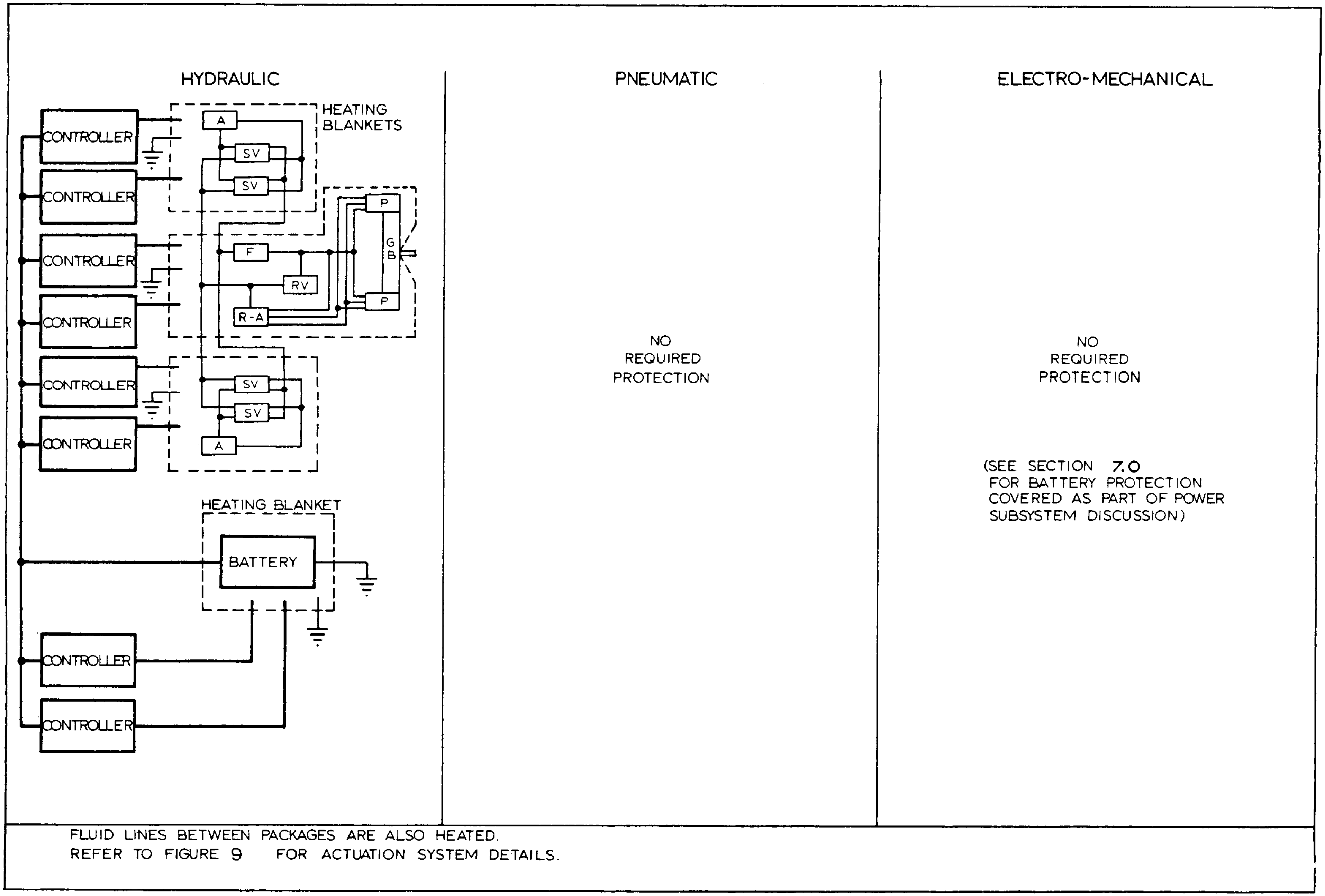


Figure 13: Thermal Protection For Lunar Landing TVC Actuation Systems

heat to maintain minimum fluid operating temperature. Before each of these periods, the conditioning heaters are required to preheat the actuation system to -100°F. During the major portion of the mission, the thrust vector control system is inactive and, with the exception of preheat periods, will be allowed to cool to the surrounding space temperature.

Actuation equipment will be packaged in three units — two servoactuators and one power assembly consisting of the gearbox, pumps, reservoir, filter, and relief valve. Each unit will be enclosed and insulated with approximately 0.1 inch of multilayer insulation. The heater will be contained within the enclosure and will be activated for 2 hours for each preheat to achieve temperature equilibrium (90 percent equilibrium is obtained in 1 hour). An additional 4-hour hold at equilibrium before usage is also included. The 3 feet of fluid lines between the power assembly and each actuator will also be insulated and will incorporate a heater.

A silver-zinc battery will be used; it will be capable of two preheats from -240°F. The battery must be insulated and heated to maintain it at 80°F throughout the 7-day mission. With 0.4 inch of insulation, the heat loss is 8.4 watt-hours per pound of battery. Insulation weight is 0.03 pound per pound of battery. The battery must heat 22 pounds of actuation equipment. The energy for one preheat from the battery was determined as follows:

1) Equipment preheating (2)

$$E = WC\Delta T$$

where

E = Total energy, Btu

W = Mass heated, lbs

C = Average specific heat, Btu/lb-°F

ΔT = Temperature increase, °F

$$E = WC\Delta T = (22)(0.1)(240-100) = 308 \text{ Btu or } 91 \text{ watt-hours}$$

where

$C = 0.1$ is assumed as an average for all equipment.

2) Heat loss during preheat and standby

During the 2-hour preheat and the 4-hour standby, heat is conducted and radiated from the equipment. Assuming a total conduction loss of 25 watts (10 watts for each actuator and 5 watts for the power unit), the 6-hour loss is 150 watt-hours. With 0.1 inch of multilayer insulation the radiation loss is 0.1 watt per square foot of surface area. With a total surface of 3.8 square feet (0.9 square feet for each actuator and 2.0 square feet for the power unit), the 6-hour loss is 3 watt-hours. The total loss during the preheat and standby is 153 watt-hours.

A battery capacity of 45 watt-hours per pound, based on withdrawing the majority of the energy in two short intervals (90 watt-hours per pound is used for slow continuous discharge). The battery must include 8.4 watt-hours per pound for self-conditioning during the 7-day mission. These considerations establish the battery requirements at

299 watt-hours (244 watt-hours for heating, 51 watt-hours for battery heating) and a weight of 6.6 pounds for one preheat. For two preheats, the battery will weigh 13.2 pounds and deliver 600 watt-hours.

Following is a description of the major conditioning system components.

1) Battery — A 28-volt silver-zinc battery providing 600 watt-hours of energy is required. The battery will be discharged in two equal periods of approximately 6 hours. These periods will be separated by a minimum of 2 days. The battery is to be provided with internal redundant heater circuits to maintain battery temperature at 80°F during a space flight of about a week. Multilayer insulation, 0.4 inch thick, will be provided for the battery.

2) Electric Heater Blankets — Redundant resistance heating blankets of different capacities and sizes are required. These are to be made of material that can withstand nonoperating soak temperatures of -240 and 200°F. Blanket requirements are:

	Size	Watts (28 Volts d.c.)
Actuactor package heater, four required	10 by 1.5 inches	35 each
Pump package heater, two required	10 by 4.5 inches	120 each
Line heater, four required	36.0 by 1.0 inch	10 each

3) Temperature Controllers — Temperature control switches are required for the electric heaters. These are small snap switches that directly switch the electricity off at -100°F and on at lower temperatures.

The weight summary for the thermal conditioning system is as follows.

	Pounds
Battery with heater, 28 volts, 600 watt-hours, one required	13.5
Battery insulation, 0.4 inch thick, 4.4 square feet	0.4
Actuation equipment insulation 0.1 inch thick, 3.5 square feet	0.2
Fluid line heater, four required, 36 by 1 inches	0.8
Actuation package heater, four required, 10 by 1.5 inches	0.4
Power assembly heater, two required, 10 by 4.5 inches	0.6
Temperature control switches, 12 required	2.4
Total	18.3

Requirements for protection of the actuation system against the hard-vacuum environment of space during the 7-day mission are not severe. Each servoactuator is capable of withstanding hard vacuum for this time without significant fluid leakage. Therefore, no vacuum protection is required for the two actuators. The pump shaft seal is a critical item with unknown performance in vacuum. It is, therefore, assumed that the power assembly is protected from vacuum with a pressurized container.

6.3.2 PROTECTION FOR PNEUMATIC CONCEPT

The pneumatically operated actuation system uses cold hydrogen gas from the RL-10 engine during operation. The system is capable of operation effectively at both -240 and 100°F. Environmental conditioning is not required, for either thermal or hard-vacuum protection (see Figure 13).

6.3.3 PROTECTION FOR ELECTROMECHANICAL CONCEPT

With respect to temperature, the motor is the critical item in the electromechanical system. The motor can be started under no load in the -240°F environment and after it has warmed can be operated under load in the -240°F environment. Cooling is not required in the 100°F environment. The battery that supplies power for the system must be equipped with an internal heater (see Figure 13). Insulation on the battery is needed to reduce losses. The amount of chargeable weight to the conditioning system is about 1 pound.

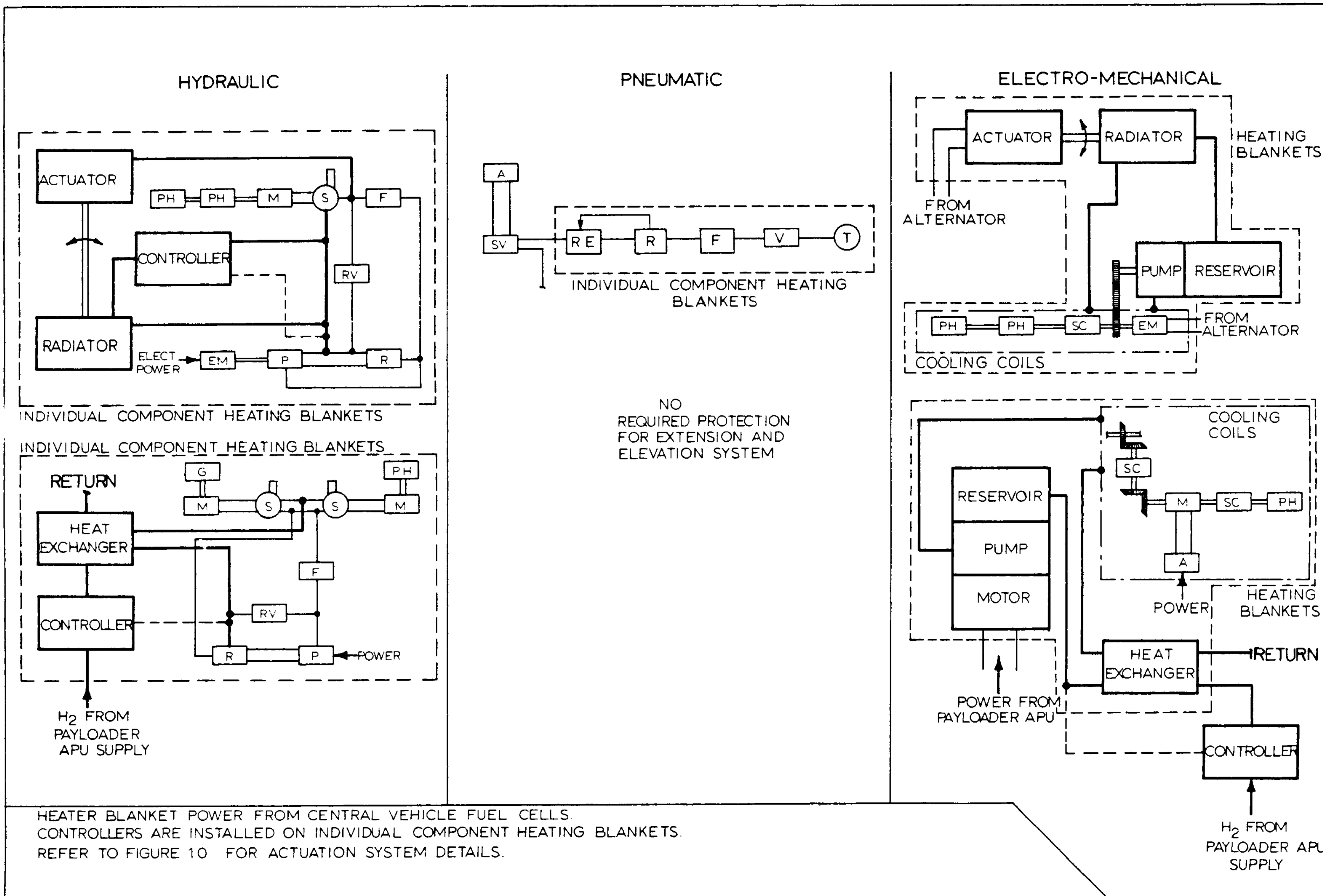
Alternating-current motors were selected for this application to eliminate the need for hard-vacuum-protection devices. None of the other components in the system is sensitive to vacuum environment.

6.4 PAYLOADER CONTROL SYSTEM

6.4.1 PROTECTION FOR HYDRAULIC CONCEPT

The thermal protection subsystem for the payloader actuation task must include provisions for both heating the system during minimum-temperature conditions and for cooling the system during the maximum temperatures. Two separate conditioning systems are required; one protects the equipment for extension and elevation of the boom and the other protects the bucket actuation equipment. Figure 14 shows schematics of these systems. Because the payloader is used intermittently, the heating for conditioning will be provided by preheating of equipment when the environmental temperature is less than -150°F. This temperature is the minimum operational temperature for the DuPont E-3 fluid in the system. Conditions requiring heating exist 50 percent of the time.

The maximum environmental temperature of 275°F when coupled with the internal heat generation of the hydraulic system establishes a requirement for cooling provisions. Cooling of the extension and elevation equipment will be provided by transferring heat to the hydrogen supplied to the power subsystem. The small power losses of the bucket actuation system will allow a small radiator to be used for this heat rejector.



HEATER BLANKET POWER FROM CENTRAL VEHICLE FUEL CELLS.
CONTROLLERS ARE INSTALLED ON INDIVIDUAL COMPONENT HEATING BLANKETS.
REFER TO FIGURE 10 FOR ACTUATION SYSTEM DETAILS.

Figure 14: Thermal Protection For Lunar Rover Payloader Actuation Systems

A radiator was selected to eliminate the need for cooling lines that would be capable of extension or retraction as the boom is operated.

The preheat function will be accomplished with electrical resistance heaters. Preheat will require approximately 40 minutes at minimum environmental temperature and will provide heating to a -100°F temperature to ensure that all fluid is not less than -150°F. Preheating time will decrease with increasing environmental temperature; at temperatures above -100°F, no preheating is required. Some of the components will be insulated to reduce the heat loss during preheat and ensure that minimum temperatures are maintained when the system is in a standby mode. It is assumed that the multipurpose vehicle to which the payloader is attached will have two 2-kilowatt fuel cells and that 3.1-kilowatts will be available for preheating. It is desirable to start preheating all equipment at the same time so the power will be allotted to the equipment to accomplish this. The fuel penalty to operate the fuel cells during preheat is discussed in Section 7.2.

6.4.1.1 Heating Requirements

A breakdown of the heating requirement is shown below. The total actuation system weight is 97 pounds; 78.5 pounds for the boom extension and elevation subsystem, and 18.5 pounds for the bucket actuation subsystem. The insulation criteria used to determine the insulation required is that the heat loss should be limited to half of the power level at minimum temperature.

1) Heating for the boom extension and elevation actuation subsystem:

    a) Insulation is not required to restrict the steady-state heat dissipation to half the average power level when the fluid is at -100°F and the environment is at -240°F;

    b) Of the total weight of the actuation system of 78.5 pounds, 69.4 pounds is equipment weight and 9.1 pounds is E-3 fluid weight. Assuming an average specific heat of 0.1 Btu/lb-°F for the equipment and 0.3 Btu/lb-°F for the fluid, the total heat from -240 to -100°F is 1350 Btu. A 50-percent allowance is assumed to account for conduction and radiation losses during the preheat, giving 2025 Btu total.

2) Heating for the bucket actuation subsystems:

    a) A 0.1-inch thickness of multilayer insulation assuming an outer surface emissivity of 0.9 (which will allow for surface degradation by lunar dust) will restrict the steady-state heat dissipation to half the average power level, with the fluid at -100°F and the environment at -240°F;

    b) Of the total weight of the actuation system of 18.5 pounds, 16.7 pounds is equipment weight and 1.8 pounds is E-3 fluid weight. Assuming an average specific heat of 0.1 Btu/lb-°F for the equipment and 0.3 Btu/lb-°F for the fluid, the total heat from -240 to -100°F is 310 Btu. A 50-percent allowance is assumed to account for conduction and radiation losses during the preheat, giving 465 Btu total.

3) Heating for the power subsystem: The combustor, turbine, gearbox, speed control, and radiator require preheating before operation.

   a) A 0.1-inch thickness of insulation on the back side of the power source radiator will reduce the preheat losses to an average of 25 Btu hr. A 0.5-inch thickness of insulation will be applied to the speed controller to ensure temperature stability during environmental fluctuations;

   b) Of the total weight of 258 pounds for the power subsystem, 108 pounds is hydrogen, oxygen, and tankage. The remaining 150 pounds are 143 pounds equipment and 7.0 pounds E-3 fluid. Assuming an average specific heat of 0.1 Btu/lb-°F for the equipment and 0.3 Btu/lb-°F for the fluid. The total heat from -240 to -100°F is 2300 Btu. A 50-percent allowance is assumed to account for conduction and radiation losses during the preheat, giving 3450 Btu total.

4) Heating for thermal protection equipment:

   Of the total thermal protection subsystem weight of 28.3 pounds, 26.5 pounds is equipment and 1.8 pounds is E-3 fluid weight. Assuming an average specific heat of 0.1 Btu/lb-°F for the equipment and 0.3 Btu/lb-°F for the fluid, the total heat from -240 to -100°F is 447 Btu. A 50-percent allowance is assumed to account for conduction and radiation losses during the preheat, giving 670 Btu total.

6.4.1.2 Cooling Provisions

Cooling provisions are required to maintain the temperature of the actuation equipment less than the limitation of the DuPont E-3 fluid (150°F when used in a high-pressure pumping system). The pumps and reservoirs will use coatings with low solar absorptivity and high emittance to limit maximum temperatures to 150°F. With these conditions the pump may be started without precooling the fluid. A breakdown of the cooling requirements is shown below.

1) Cooling of the boom extension and elevation actuation system:

   a) The boom extension and elevation system is cooled by transferring heat to the hydrogen supplied to the power subsystem. Temperature control of the hydraulic fluid will be accomplished by the hydrogen bypassing the heat exchanger. The temperature control will be set to provide full hydrogen bypass when the hydraulic fluid outlet temperatures reach 0°F and proportional control flow to full open at hydraulic outlet temperatures of 50°F;

   b) The energy to be dissipated through the hydraulic fluid to the hydrogen heat exchanger includes the pump losses of 3640 Btu/hr and the solar heating of approximately 1400 Btu/hr.

   c) At maximum heat load, hydrogen enters the exchanger at a maximum temperature of -340°F and increases to -100°F at the exit while flowing at the rate of 5.7 pounds per hour. The E-3 fluid temperature is decreased from 100 to 50°F. This requires one square foot of heat transfer area in the heat exchanger.

2) Cooling of the bucket actuation subsystem:

   a) The bucket rotation subsystem is cooled with a small radiator located on the boom near the bucket. The radiator includes provisions for the operator to remotely position the radiator. This feature allows the system to be designed for a sink temperature of 65°F. Hydraulic fluid will circulate continuously through the radiator. The fluid thermal inertia will allow the radiator to be sized for average power dissipation;

   b) The radiator surface exposed to the lunar surface is protected with a 0.5-inch thickness of multilayer insulation to reduce the heat gain from a hot lunar surface to 2.4 Btu/hr.

   c) The radiator is sized for 125 Btu/hr with a sink temperature of approximately 65°F. This requires two square feet of radiator surface.

3) Cooling for the power subsystem:

   The power subsystem uses an active radiator to reject the gearbox and turbine losses. The circuit uses E-3 as the coolant fluid. This system is a part of the power subsystem described in Section 7.2.

4) Cooling for the thermal protection equipment:

   No special cooling provisions are required for the components of the thermal protection system.

6.4.1.3 Hydraulic System Major Conditioning Components

A description of the major hydraulic system conditioning components follows.

1) Hydraulic fluid to hydrogen heat exchanger and controller:

   a) A parallel-cross-flow (shell and tube) heat exchanger and mechanical thermal control valve are required for a maximum heat exchange of 5000 Btu/hr and minimum exchange of 500 Btu/hr.

   b) At maximum heat exchange the hydraulic fluid conditions are:

      Inlet temperature: 100°F

      Outlet temperature: 50°F

      Flow rate: 392 pounds per hour

      ΔP (allowable): 20 psi

   c) For the same conditions, the hydrogen heat transfer fluid conditions are:

      Inlet temperature: -340°F

      Outlet temperature: -100°F

      Flow rate: 5.7 pounds per hour

      ΔP (allowable): 5 psi

d) At minimum heat load, a controller maintains the hydraulic temperature at no less than 0°F by bypassing hydrogen flow. A mechanical thermal controller is used.

2) Electric heater blankets:

The following electrical resistance heating blankets are required. These must be made of material that can be cold-soaked at -240°F and then operate. They must withstand temperatures in the nonoperating condition to 275°F, except the special heaters noted that must be capable of 500°F as a nonoperating condition.

	Size (inches)	Area (square inches)	Power (watts)
Boom extension heaters:			
Gearbox	6.0 by 6.0	36.0	150
Motor	2.5 by 4.5	11.2	25
Solenoid valve	1.0 by 3.0	3.0	30
Boom elevation heaters:			
Hinge, two required	6.0 by 3.0	36.0	350 each
Motor	2.5 by 4.5	11.2	25
Solenoid	1.0 by3.0	3.0	30
Filter	4.0 by 1.0	4.0	40
Relief valve	6.0 by 1.0	6.0	15
Pump	3.0 by 5.0	15.0	160
Reservoir	3.0 by 5.0	15.0	150
Tubing, eight required	1.0 by 15.0	120.0	10 each
Bucket elevation heaters:			
Hinge, two required	2.0 by 3.0	12.0	65 each
Motor	2.5 by 4.5	11.2	20
Solenoid	1.0 by 3.0	3.0	30
Relief valve	6.0 by 1.0	6.0	15
Filter	0.5 by 1.0	0.5	2
Pump	2.5 by 4.5	11.2	20
Reservoir	2.5 by 4.5	11.2	15
Motor	4.5 by 4.5	20.0	110
Tubing, four required	1.0 by 15.0	60.0	10 each

	Size (inches)	Area (square inches)	Power (watts)
Power subsystem:			
Gearbox (500°F heater required)	2.0 by 6.0	12.0	60
Turbine (500°F heater required)	2.0 by 10.0	20.0	110
Combustor (500°F heater required)	3.0 by 5.0	15.0	65
Speed Control	10.0 by 5.0	50.0	75
Radiator, four required	6.0 by 48.0	1152.0	280 each
Thermal protection subsystem:			
Heat exchanger and valve	4.0 by 4.0	16.0	150
Bucket elevation radiator	12.0 by 24.0	288.0	100
	Total	1950	3467

Total estimated weight: 5.4 pounds (Based on 0.4 pound per square foot)

3) Temperature controllers:

Temperature control switches are required for the electric heaters. These are small snap switches that directly switch the power off at -100°F and on at lower temperature.

4) Radiator:

The radiator, 2 feet by 1 foot, consists of a plate with 3/16-inch tubes at 2-inch spacing. The radiator is mounted on two bearings that allow it to be rotated in a 120-degree arc. The actuator for the radiator is integral with the radiator. The actuator uses DuPont E-3 fluid in the bucket actuation system and is controlled by a solenoid valve. Radiator positioning is controlled remotely by the payloader operator. Thermal cycling tests of a DuPont E-3 hydraulic system reported in Volume II show that freezing the fluid will not affect its performance.

6.4.1.4 Thermal Conditioning System Weight Summary

A weight summary for the thermal conditioning system is as follows.

	Pounds
Boom extension and elevation subsystem:	
Exchanger	3.0
Controller	1.0

	Pounds
Fluid	1.5
Heaters	0.8
Heater temperature controllers	4.0
Bucket rotation subsystem:	
Insulation	0.4
Radiator (includes 0.6 pound of insulation)	3.6
Radiator position actuator	1.0
Fluid	0.3
Heaters	1.3
Heater temperature controllers	2.8
Power subsystem:	
Insulation (4.0 pound for radiator and 0.3 pound for speed control)	4.3
Radiator	*
Fluid	*
Heaters	3.3
Heater temperature controllers	1.0
Total	28.3

*Included in power source weight summary.

The amount of protection for operation in hard vacuum is determined to be insignificant on the basis of vacuum storage tests reported in Volume II. However, the hydraulic boom actuation equipment is considered to be compartmented at the boom base so that cabin pressurization can be supplied to it, if desired to improve reliability.

Bucket actuation equipment is also considered to be encapsulated with a dust cover that can be pressurized, if desired.

6.4.2 PROTECTION FOR THE PNEUMATIC CONCEPT

There are no known thermal protection requirements for the hydrogen-oxygen pneumatic system used to actuate the extension-elevation boom. The hydrogen and oxygen tanks will either not be installed or will be emptied when the payloader is inactive.

The hydrazine-gas-generation portion of the bucket actuation system requires preheat conditioning for environmental temperatures below -65°F. The heat generated in the decomposition chamber will keep the equipment warm during operation periods (see Figure 14). Preheat conditioning will keep the hydrazine from freezing in valves or at the inlet to the decomposition chamber. Conditioning of the hydrazine tank might not be necessary if the tank is installed immediately before use. The conservative design approach of providing heating was adopted.

Preheating will be provided in the same manner as provided with the hydraulic concept. Twelve watts of power from the central roving vehicle fuel cells are required to provide preheating. This amount of power is considered negligible, so a fuel penalty to operate the cells was not calculated.

A description of the conditioning components follows:

	Area (square inches)	Power (watts)	Weight (pounds)
Bucket elevation heaters			
Hydrazine tank	109	8	1.2
Decomposition chamber plumbing and controls	162	4	0.6
Temperature controllers, two required	--	-	0.4
Insulation			
Hydrazine tank	109	-	0.1
Decomposition chamber plumbing and controls	162	-	0.1
	Total		2.4

The protection considerations for operation of the pneumatic system in a hard-vacuum environment are considered to be negligible. Components are packaged to prevent dust contamination and to permit pressurized environment, if necessary.

6.4.3 PROTECTION FOR THE ELECTROMECHANICAL CONCEPT

The thermal protection subsystem for the payloader actuation task must include provisions for heating the system during minimum temperatures and cooling the system during maximum temperatures. Two separate systems are required, one for extension and elevation of the boom and one for rotating the bucket. Figure 14 shows these systems schematically.

The electromechanical actuators can be operated in the minimum environmental temperature of -240°F by starting the a.c. motors at zero load and allowing heat losses to warm the unit. By using this method, a preheat of the actuation subsystem is not required. Both the power and thermal protection subsystems require preheat or must be maintained above their minimum operating temperatures. Because of the possible extended period of inoperation of the payloader, the concept selected for the trade study is to allow the components to cycle with the environmental temperature and provide a suitable preheating function.

The preheat, when required, will be accomplished with electrical resistance heaters. The fuel cell of the multipurpose vehicle will provide the electrical power for the preheat. The multipurpose vehicle is assumed to have 4 kilowatts of electrical power of which 3.1 kilowatts can be available to preheat the active conditioning system and power subsystem. The preheat will require about 30 minutes when the components are at minimum environmental temperature. The preheating time will decrease with increasing environmental temperature; at temperatures above -100°F, no preheating is required. Some of the components will also be insulated to reduce the heat loss during preheat and ensure that minimum temperatures are maintained when the system is in standby. The fuel penalty to operate the fuel cells during the preheat is discussed in Section 7.2.

The maximum environmental temperature of +275°F with the strenuous duty cycle will result in temperatures that exceed the maximum continuous operating temperature of electromechanical components; therefore, cooling must be provided. Because the electromechanical equipment will operate satisfactorily at 275°F, no precooling before starting is required.

The boom extension and elevation system is cooled by transferring heat to the hydrogen supplied to the power subsystem. A heat transport loop using DuPont E-3 fluid will transfer the heat from the actuation system components to the hydrogen.

The small power losses of the bucket rotation system will allow a small radiator to be used for heat rejection. By using a separate radiator located near the bucket, only electrical wires are required to connect the bucket actuation system with the lower chassis of the payloader.

6.4.3.1 Heating Requirements

A breakdown of the heating requirement is given below.

1) Heating for the power subsystem:

   The combustor, turbine, gearbox, alternator, and radiator require heating to -100°F, and the speed controller to 0°F before operation.

   a) A 0.1-inch thickness of insulation is installed on the back side of the 70 square foot power source radiator to reduce the heat loss during preheat to an average of 25 Btu/hr. A 0.5-inch thickness of insulation will be applied to the speed controller to ensure temperature stability during environmental fluctuations.

   b) Of the total weight of 269 pounds for the power subsystem, 109.2 pounds are hydrogen, oxygen, and tankage. Of the remaining 159.8 pounds, 152.8 pounds are equipment and 7.0 pounds are DuPont E-3 fluid. Assuming an average specific heat of 0.1 Btu/lb-°F for the equipment and 0.3 Btu/lb-°F for the fluid, the total heat from -240 to -100°F is 2430 Btu. A 50-percent allowance is assumed to account for conduction and radiation loss during the preheat, giving 3650 Btu total.

2) Heating for the thermal protection subsystem:

The components of the thermal protection subsystem will require preheating before the actuation subsystem can be started in the -240°F environment. This is necessary to melt the DuPont E-3 heat transport fluid used to cool the alternator. To ensure that all of the DuPont E-3 is in a liquid state, the motors, spring clutches, and power hinges of the actuation components will also be heated. Approximately 150 pounds of the actuation subsystem and thermal protection subsystem will be warmed to ensure the 5.7 pounds of DuPont E-3 is melted. Assuming an equipment average specific heat of 0.1 Btu/lb-°F and fluid specific heat of 0.3 Btu/lb-°F the total heat from -240 to -100°F is 2340 Btu.

6.4.3.2 Cooling Provisions

Both the boom and bucket actuation subsystems use heat transport loops to transfer the energy from the electromechanical components to the sink. The sink used for the boom extension and elevation subsystem is the hydrogen used for fuel by the power source; heat from the bucket rotation subsystem is radiated to the environment.

Both the boom and bucket heat transport circuits will use DuPont E-3 fluid. This fluid operates from -150 to +300°F when used in a relatively low-head positive-displacement pumping system.

A breakdown of the cooling requirements is shown below:

1) Cooling for the boom extension and elevation actuation system — Total heat to be dissipated is:

   a) Actuation components cooling load:

Peak loss	9660 Btu/hr
Average loss	2257 Btu/hr;

   b) Alternator cooling load:

Peak loss	7005 Btu/hr
Average loss	1850 Btu/hr
Total dissipation	16,665 Btu/hr (peak) 4107 Btu/hr (average);

   c) The energy is dissipated from the DuPont E-3 heat transfer fluid to the hydrogen heat sink in a cross-parallel flow, shell and tube, heat exchanger sized for 10,000 Btu/hr. The exchanger is adequate to handle the transient peak loads of 16,665 Btu/hr;

   d) At maximum heat load the hydrogen will enter the exchanger at a maximum of -340°F and exit at 140°F flowing at the rate of 5.7 pounds per hour. The DuPont E-3 fluid temperature will be decreased from 300 to 250°F. Because of the proximity of the DuPont E-3 fluid operating temperature to the maximum environmental temperature, no solar heat load or conduction load from adjacent structure is assumed. This requires 2 square feet of heat transfer area in the exchanger.

2) Cooling for the bucket actuation subsystem:

The bucket rotation subsystem is cooled by a small radiator, located on the boom near the bucket. The heat transport fluid will circulate continuously through the radiator. The thermal capacitance of the fluid and system will allow the radiator to be sized for average power conditions.

a) The radiator will be sized for 100 Btu/hr with a sink temperature of 65°F;

b) One-half square foot of radiator area is required to maintain an average radiator surface temperature of 260°F and maintain the radiator outlet maximum fluid temperature of 250°F.

3) Cooling for the power subsystem:

The power subsystem will use an active radiator to reject the gearbox and turbine losses. The circuit will utilize DuPont E-3 as the heat transfer fluid. This system has been included as part of the power subsystem and is described in Section 7.2.

4) Cooling for thermal protection equipment:

No special cooling provisions are required for the components of the thermal protection system.

6.4.3.3 Electromechanical System Major Conditioning Components

A description of the major electromechanical system conditioning components is as follows:

1) DuPont E-3 to hydrogen heat exchanger and controller:

a) A parallel-cross-flow (shell and tube) heat exchanger and mechanical thermal temperature control valve are required for a minimum heat exchange of 500 Btu/hr and maximum exchange of 10,000 Btu/hr;

b) At maximum heat exchange the DuPont E-3 fluid conditions are:

Inlet temperature: 300°F
Outlet temperature: 250°F
Flow rate: 835 pounds per hour
ΔP (allowable): 10 psi;

c) For the same conditions the hydrogen heat-sink conditions are:

Inlet temperature: -340°F (maximum)
Outlet temperature: 140°F
Flow rate: 5.7 pounds per hour
ΔP (allowable): 5 psi;

d) At minimum heat load, a controller maintains the heat transport fluid temperature at no less than 100°F bypassing hydrogen flow. A mechanical thermal controller is used.

2) Electric heater blankets:

The following electrical resistance heating blankets are required. These must be made of material that can be cold-soaked at -240°F and then operated. They must withstand temperatures in the nonoperating condition to 300°F, except the special heaters noted that must be capable of 500°F as a nonoperating condition.

	Size (inches)	Area (square inches)	Power (watts)
Boom actuation cooling unit:			
Heat exchanger	3 by 7	21	85
Pump and reservoir	5 by 10	50	250
Alternator	5 by 10	50	250
Electromechanical motor	5 by 10	50	250
Clutch-gearbox assembly, two required	7 by 7	100	125 each
Lines, ten required	1 by 15	150	10 each
Bucket actuation cooling unit:			
Radiator	12 by 6	72	20
Pump and reservoir	5 by 5	25	135
Electromechanical motor	2 by 5	10	30
Power hinges, two required	2 by 3	12	40 each
Lines, five required	1 by 15	75	10 each
Power subsystem:			
Gearbox (500°F heater)	2 by 6	12	150
Turbine (500°F heater)	2 by 10	20	200
Combustor (500°F heater)	3 by 5	15	150
Speed control and voltage regulator	10 by 5	50	160
Radiator, four required	6 by 48	1152	375 each
Totals		1864	3660

3) Temperature controllers:

Temperature control switches are required for the electric heaters. These are small snap switches that directly switch the power off at -100°F and on at lower temperature.

4) E-3 fluid pump and reservoirs:

Two positive-displacement gear pumps with electrical drive and reservoirs are required. Detailed requirements for each pumping unit are:

a) Boom actuation cooling system pumping unit: 1-gpm output from an electric-driven motor-pump with 80 watts peak input. Reservoir capacity is 50 cubic inches;

b) Bucket actuation cooling system pumping unit: 0.1-gpm output from an electric-driven motor-pump with 50 watts peak input. Reservoir capacity is 25 cubic inches.

5) Boom system radiator:

The radiator 0.5- by 1-foot consists of a plate with 3/16-inch tubes at 2-inch spacing. This is mounted on two bearings that allow it to be rotated in a 120-degree arc. The actuator for this is part of the radiator and is electrically driven and is capable of being positioned remotely by the operator.

6.4.3.4 Thermal Conditioning Weight Summary

The weight summary for the thermal conditioning system is:

	Pounds
Boom extension and elevation subsystem:	
Exchanger	4.5
Pump and reservoir	8.0
Controller	1.0
Lines	0.5
Fluid	4.4
Heaters	1.6
Heater temperature controllers	4.0
Bucket rotation subsystem:	
Radiator	0.8
Radiator position actuator	1.0
Pump and reservoir	5.0
Fluid	1.3
Lines	0.2
Heaters	0.5
Heater temperature controllers	2.0

	Pounds
Power subsystems:	
Insulation (4.0 pounds for the radiator and 0.3 pound for the speed control)	4.3
Radiator	*
Fluid	*
Heaters	3.3
Heater temperature controllers	1.2
Total	43.6

*Included in power source weight summary.

Alternating current motors were used in the electromechanical concept to eliminate the need for vacuum environmental protection.

7.0 POWER SYSTEM DESIGNS

The evaluation of space actuation tasks is not complete unless consideration is given to the power sources necessary for operating the actuation equipment. The selection of practical power sources to operate each actuation concept considered for each of the four tasks is based on the ease of using power already available on the basic vehicles shown in Figure 3.

The power required for the hydraulic and electromechanical concepts used for elevon actuation is provided in each case by increasing the capacity of the central hydrogen-oxygen turbine power source on the re-entry vehicle. The amount of the increase is considered chargeable to the actuation system as a power penalty. Details of these penalties are discussed in Section 7.1. The pneumatically actuated elevon system is powered by solid-propellant generators installed specifically for this actuation function and entirely chargeable to the actuation system. A description of the generator requirements has been included in Section 5.1.

The power to operate the centrifuge may be considered to be available at no penalty to the actuation systems evaluated. The power magnitude is low, and it is considered that the men on the centrifuge are relieved of operating equipment that would draw a power level at least equivalent to that of the centrifuge requirement.

The RL-10 engine to be gimbal-actuated on the lunar-landing vehicle has an accessory pad used to directly power the hydraulic actuation concept. There is no associated power penalty with this application because the turbine pad is available and the ratio of accessory power to main-engine power is insignificant. This pad provides a maximum of 2.5 horsepower at 11,300 rpm. The pneumatic gas pressure for TVC actuation is also provided at insignificant penalty by bleeding hydrogen from the RL-10 turbine inlet. Details of the hydrogen bleed have been discussed in Section 5.3. The power for the electromechanical concept is provided by an a.c. motor-driven system using a battery/inverter, and was selected on the basis of weight.

The concept of the payloader as an attachment to a multipurpose lunar-roving vehicle might be considered to be most logically powered by increasing the capacity of the roving-vehicle-fuel-cell system and using this power for all attachments. Because the duty cycle for the payloader attachment is comprised of extreme variations in load and short durations between load spikes, fuel-cell power is not well suited to the application.

The hydraulically operated payloader uses a hydrogen-oxygen APU to supply power for the extension-elevation function; this power supply being provided only for this function. System design is detailed in Section 7.2.1. The bucket-tipping function performed by hydraulics requires approximately 0.5 horsepower at intervals when the roving vehicle is stationary and vehicle fuel-cell power may be provided to the actuation system at insignificant penalty.

The gas required for pneumatic power to operate the payloader is provided by a liquid hydrogen/liquid oxygen system for the extension-elevation function, and by a separate hydrazine system for the bucket-actuation function. These power systems have been discussed in detail in Section 5.4. The entire electromechanical payloader system is separately powered from the vehicle by a hydrogen-oxygen APU similar to the one for hydraulic power. Details of this APU design are discussed in Section 7.2.2.

7.1 ELEVON CONTROL SYSTEM

7.1.1 POWER FOR HYDRAULIC CONCEPT

The power penalty to be charged to the hydraulic actuation system for elevon control task is in the equipment that is provided on the manned re-entry vehicle for the sole purpose of satisfying actuation requirements. The shuttle-vehicle power profile with hydraulic actuation, shown in Figure 15, requires a peak power level of 3.2 horsepower to be delivered to utilities (excluding actuation equipment) throughout the vehicle. An additional 11.8 horsepower is required to supply the actuation-equipment peak demands. These peak-power requirements can be easily handled by the chemical dynamic system, shown schematically in Figure 16, using three turbine power units operating simultaneously. During the major portion of the flight, the average vehicle load is carried by only one unit. This provides a condition whereby any single turbine unit is always operating close to its design load. This operation results in a low specific fuel consumption throughout the flight, and because two units are on standby most of the time, a high reliability is obtained.

Because it was lighter, a hydrogen-oxygen turbine power system was selected in preference to a fuel-cell system. The system design assumes the characteristics of a constant combustor-outlet temperature of 1550°F, pressure ratio at maximum load of 166.6, and a combustion-pressure at maximum load of 250 psi. It is reasonable to expect that a combustor inlet temperature of 85°F can be utilized, thereby providing a combustion specific impulse of 388 seconds.

The turbine power system provides control power for only the re-entry maneuver, the boost abort power being provided by the hydraulic accumulator described in Section 5.1. The peak maneuver requirements demand that the three APU alternators provide 17.2 horsepower or 5.73 horsepower from each unit. Each turbine is required to develop 9.1 horsepower. The average total gas flow is 7.05 pounds per hour at a fuel-to-oxdizer mixture ratio of 1.16.

Heat rejection from the APU gearboxes and alternators averages 7020 Btu/hr which is used entirely to heat the hydrogen fuel to the combustion inlet temperature of 525°R. This method requires no additional hydrogen for cooling the system. A weight summary for the chargeable power penalty to the hydraulic concept is as follows.

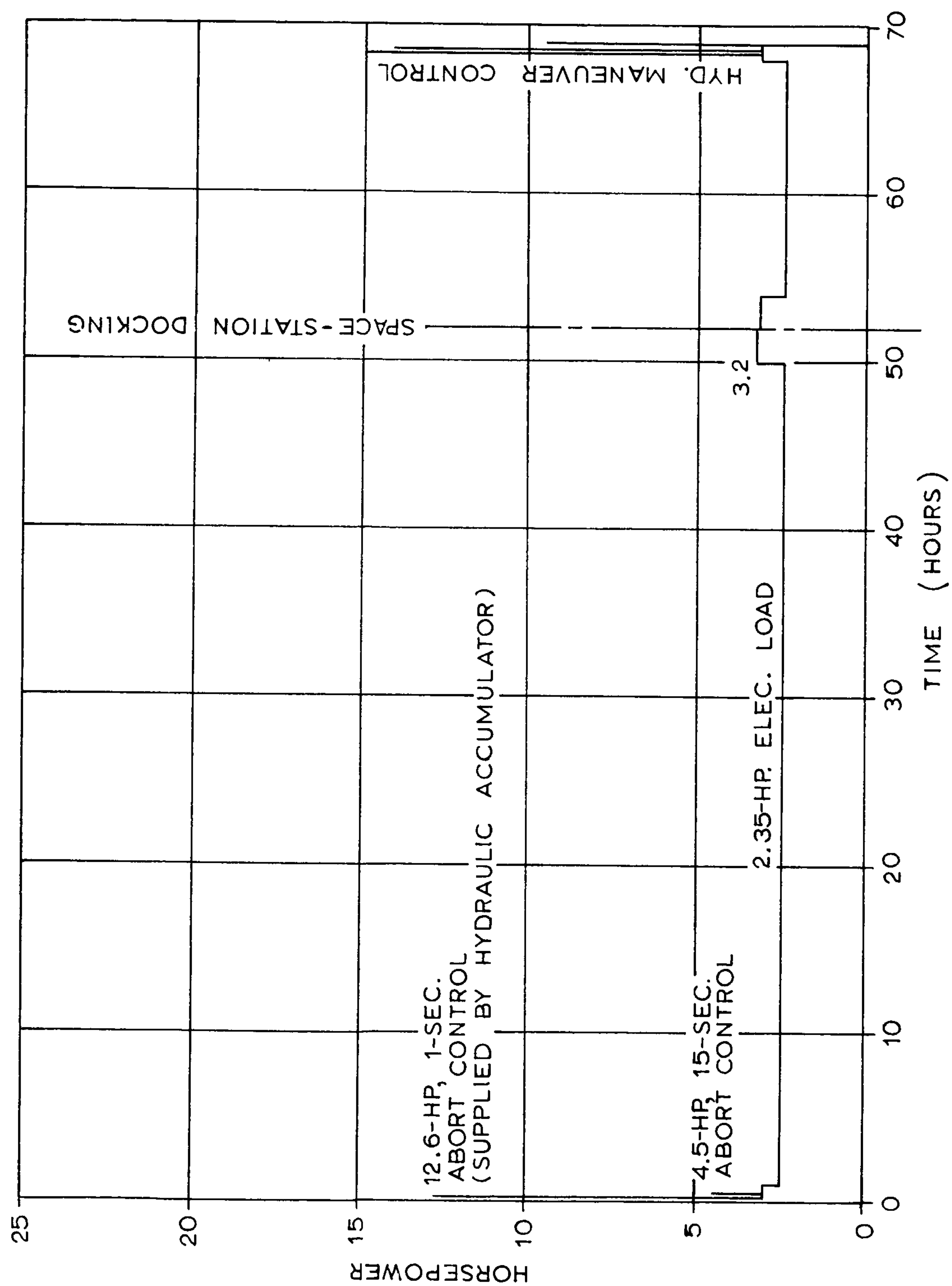


Figure 15: Power Profile For Shuttle Vehicle Hydraulic Actuation

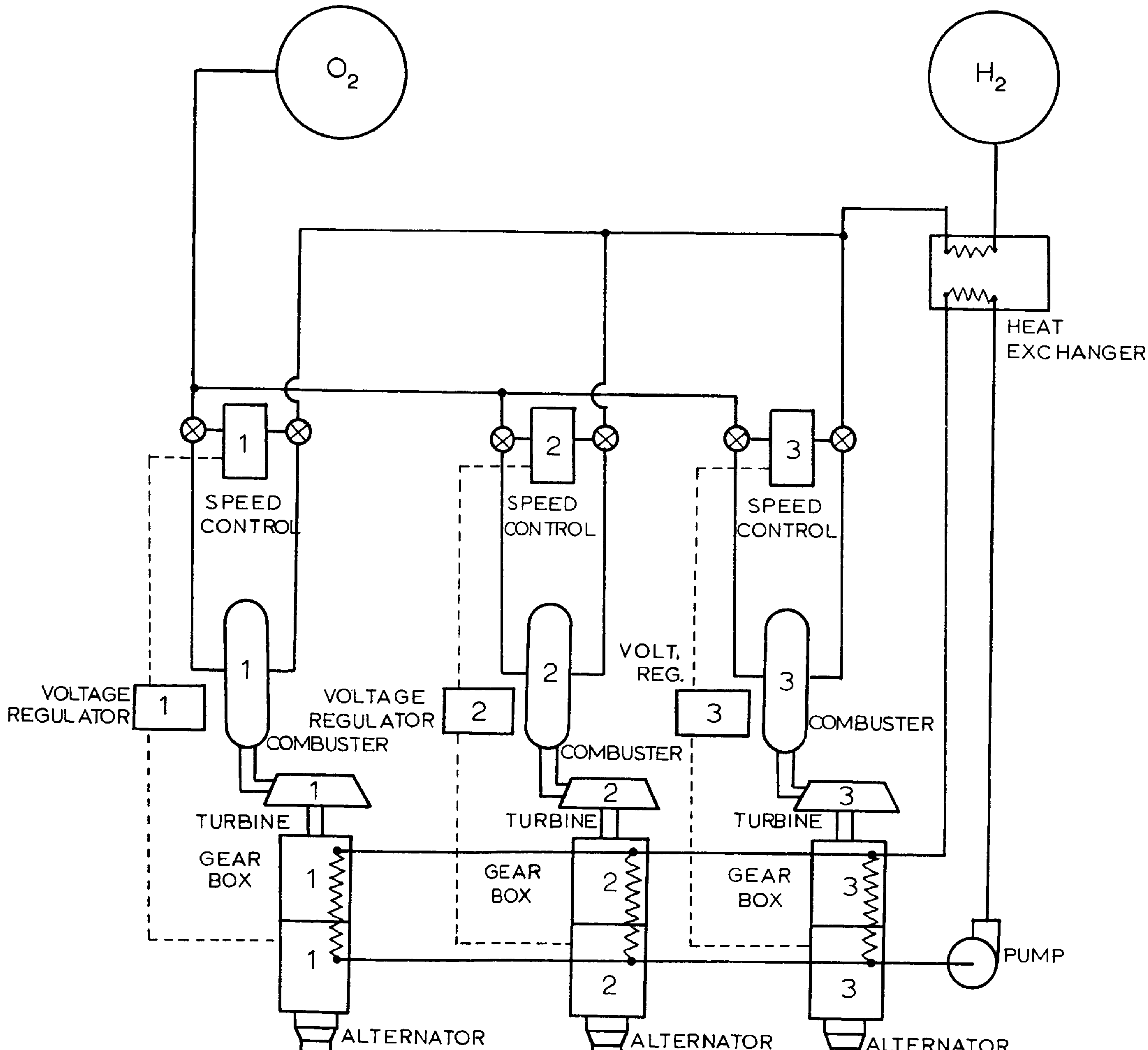


Figure 16: Shuttle Vehicle Chemical Dynamic Power System

Power System Weight Summary — Hydraulic Concept (Pounds)

	Excluding Actuation	Including Actuation	Power Penalty
Hydrogen	294.0	349.0	55.0
Hydrogen tank	250.0	290.0	40.0
Oxygen	189.0	225.0	36.0
Oxygen tank	59.0	67.0	8.0
Alternators, three required	25.8	38.4	12.6
Gearboxes, three required	4.5	6.0	1.5
Turbines, three required	8.7	16.8	8.1
Combustors, three required	4.8	21.0	16.2
Voltage regulator, three required	12.0	12.0	--
Speed control, three required	4.5	4.5	--
Brackets	2.7	3.9	1.2
Tubing, insulation, valves	85.5	103.4	17.9
Heat exchanger and pump	7.5	10.0	2.5
Heat transfer fluid	5.5	7.0	1.5
Total	953.5	1154.0	200.5

7.1.2 POWER FOR ELECTROMECHANICAL CONCEPT

The shuttle-vehicle power profile with electromechanical actuation is shown in Figure 17, and requires the same utility-power level, exclusive of actuation, as with the hydraulic concept. An additional 25 horsepower is required to supply the actuation-equipment peak demands during the abort period, with a peak demand of 6.3 horsepower during the re-entry maneuver period. Three simultaneously operating hydrogen-oxygen turbine-powered alternators, each rated at 6.2 horsepower, will satisfy these requirements assuming that intermittent 5-second operation at 150 percent of rated power may be incorporated in the design. Each turbine will be rated at 11.3 horsepower. During the major portion of the flight, the average vehicle load is 2.43 horsepower and is carried by only one unit. The system is schematically shown in Figure 16, and differs only in size from the system required to power the hydraulic concept.

The average heat-rejection rate from the APU gearboxes and alternators is 7520 Btu/hr which is used entirely to heat the hydrogen fuel to the combustion inlet temperature of 525°R. No additional hydrogen is required for cooling the system. A weight summary for the chargeable power penalty to the electromechanical concept is as follows.

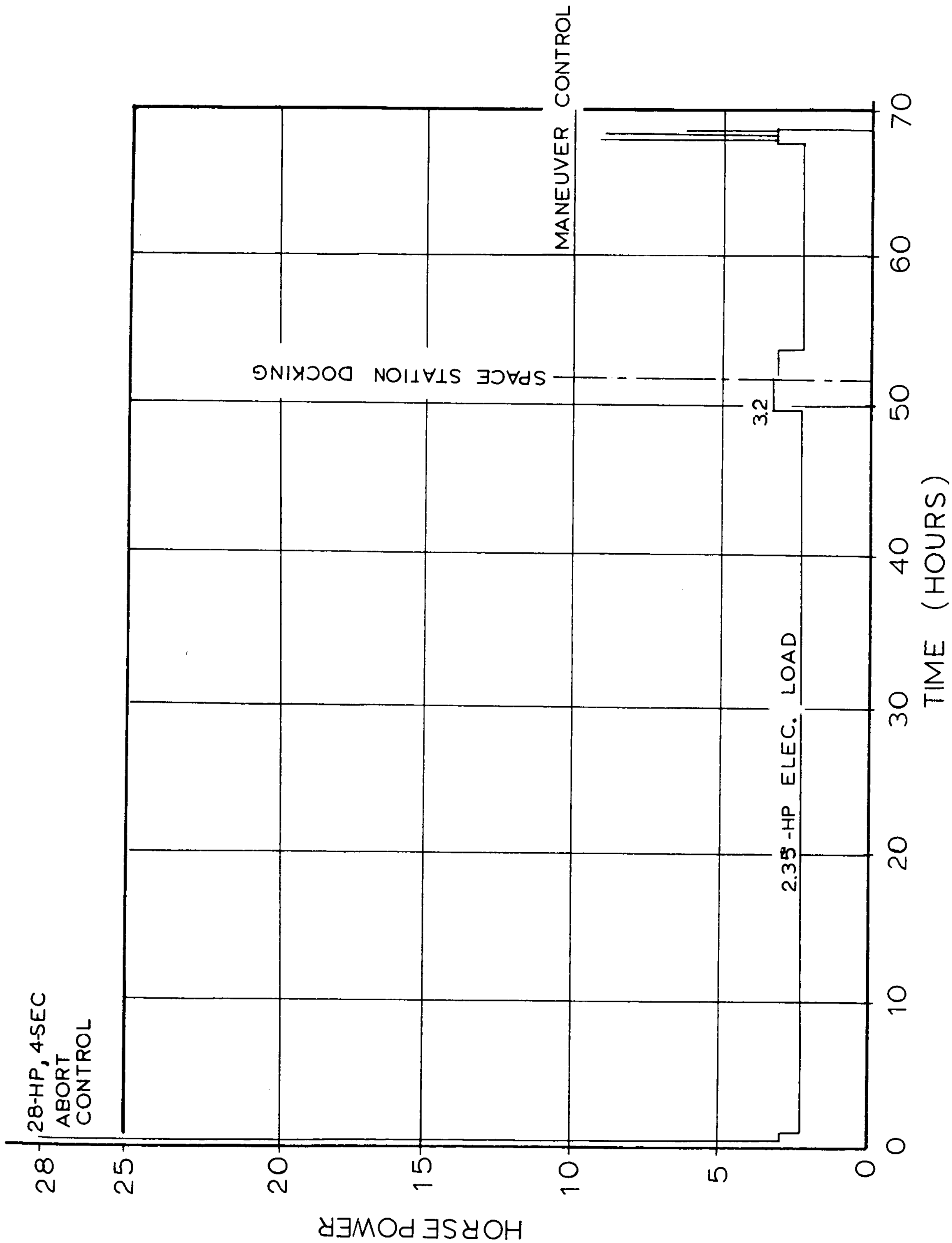


Figure 17: Power Profile For Shuttle Vehicle Electromechanical Actuation

Power System Weight Summary — Electromechanical Concept (Pounds)

	Excluding Actuation	Including Actuation	Power Penalty
Hydrogen	294.0	379.0	85.0
Hydrogen tank	250.0	305.0	55.0
Oxygen	189.0	234.0	45.0
Oxygen tank	59.0	69.0	10.0
Alternators, three required	25.8	41.7	15.9
Gearboxes, three required	4.5	9.0	4.5
Turbines, three required	8.7	20.1	11.4
Combustors, three required	4.8	25.2	20.4
Voltage regulator, three required	12.0	15.0	3.0
Speed control, three required	4.5	7.5	3.0
Brackets	2.7	4.8	2.1
Tubing, insulation, valves	85.5	111.0	25.5
Heat exchanger and pump	7.5	11.0	3.5
Heat transfer fluid	5.5	7.5	2.0
Total	953.5	1239.8	286.3

7.2 PAYLOADER CONTROL SYSTEM

7.2.1 POWER FOR HYDRAULIC CONCEPT

An accessory power unit is provided on the payloader attachment to the lunar-roving vehicle to supply all power for hydraulic extension and elevation of the payloader boom. The duty cycle for the combined boom extension and elevation function is shown in Figure 18. The turbine power unit is not allowed to operate without load; therefore, a 1 horsepower by-pass load is added to the duty cycle when hydraulic actuation power is not required. It is assumed that at the low hydraulic power output requirement of 0.25 horsepower, pump losses alone provide an adequate turbine load; hence, additional parasitic losses are not required. The average power output at the hydraulic pump is 3.58 horsepower, requiring an average turbine output of 7.5 horsepower with a design maximum of 17 horsepower. The APU requires a fuel consumption of 10.6 pounds per hour at an H_2 to O_2 ratio of 1.17.

Based on comparative weights for an expected life of approximately 20 hours of operation with refueling at 4-hour intervals, a hydrogen-oxygen turbine power system was selected in preference to a fuel cell system. Figure 19 shows a comparison of how

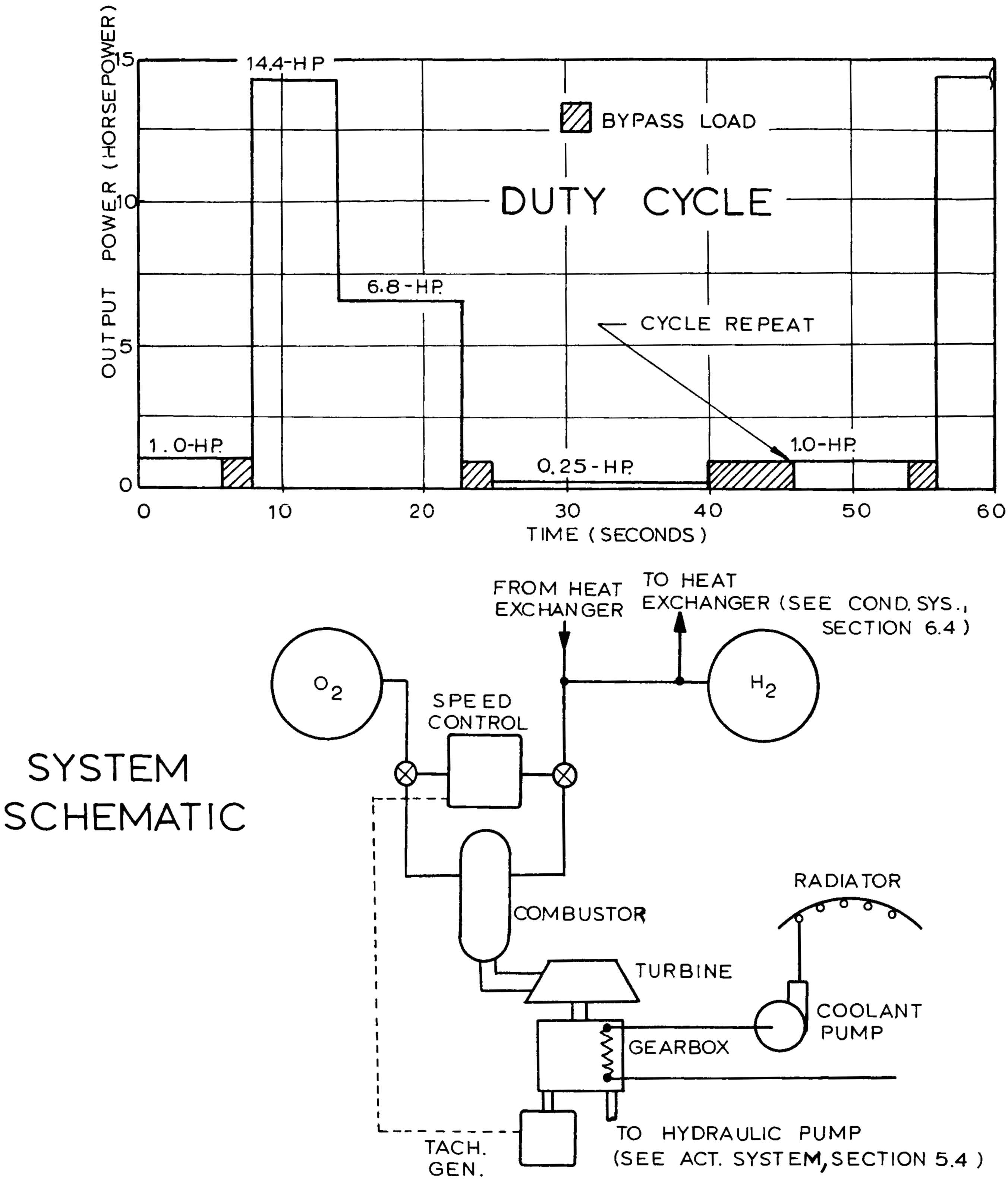


Figure 18: Payloader Actuation Power System-Hydraulic Concept

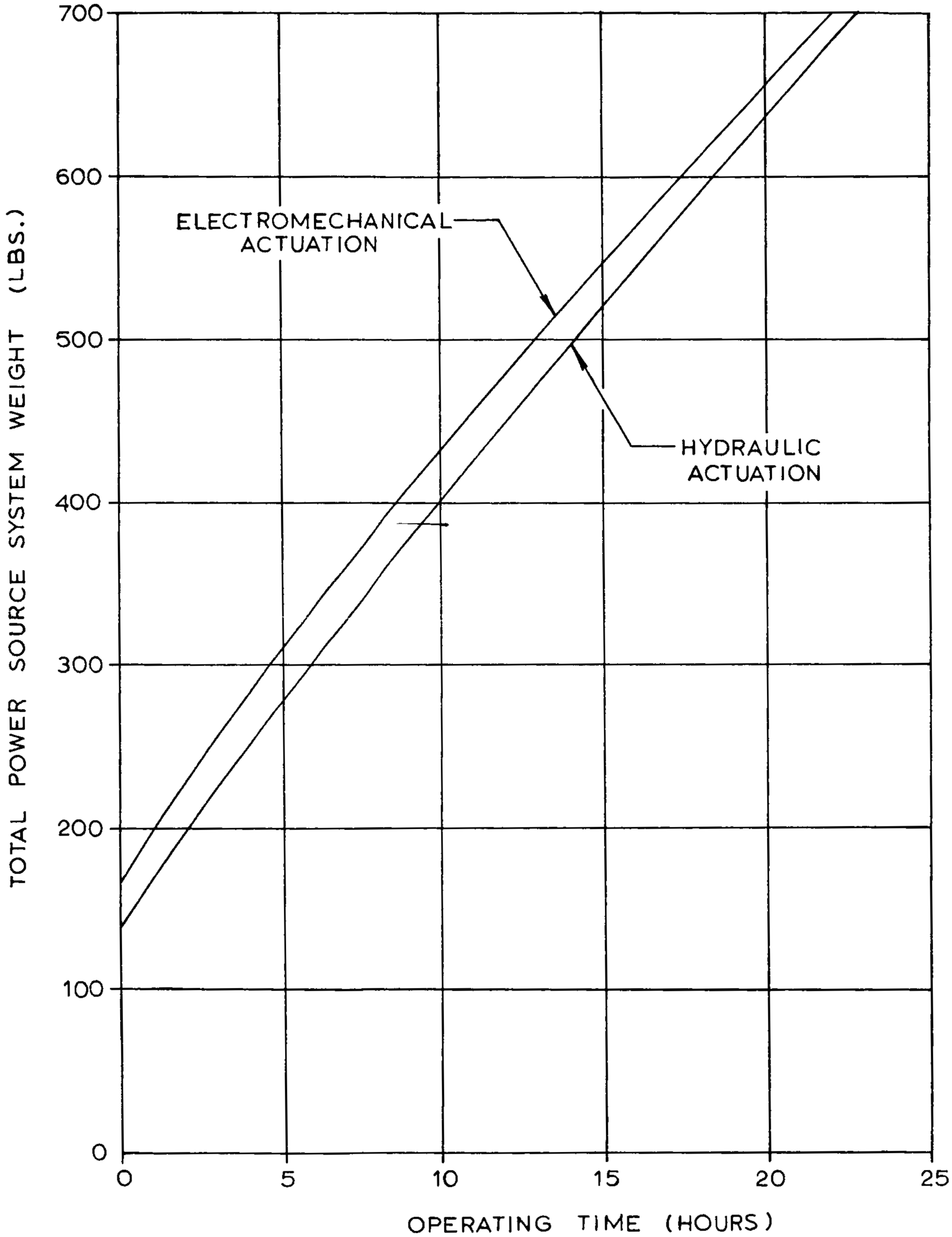


Figure 19: Payloader Power Unit-Weight Vs Operating Time

both hydraulic- and electromechanical-power-system weights increase with operation time for a single fueling. The system-design schematic for the payloader power system is shown in Figure 18, and assumes a constant combustor-outlet temperature of 1550°F and a combustion pressure at maximum load of 250 psi. The combustor-inlet temperature is 0°F, providing a combustion specific impulse of 382 seconds.

The hydrogen preheating capacity to provide the 460°R inlet temperature is 8980 Btu/hr. All the hydraulic pump losses, averaging 3640 Btu/hr, can be absorbed in preheating the hydrogen. Additional heat may be provided from the exhaust of the APU turbine. This heating could be provided from a portion of the gearbox losses, which average 7100 Btu/hr. To prevent complicating the heat transfer problem, a decision was made to reject all the gearbox losses with a radiator rather than split the losses between the radiator and hydrogen preheating.

Radiator design for gearbox cooling is based on an effective sink temperature of 65°F for the lunar surface. This value for sink temperature is used in lieu of a total mission study required to define an optimum value, such analysis being beyond the scope of this contract. Using Figure 20, a radiator size of 70 square feet was selected as near optimum. The heat-transport fluid is DuPont E-3, which enters the radiator at a temperature of 290°F.

When payloader utilization is required in a minimum lunar-surface environment, the power subsystem components need to be heated before startup. Preheating, discussed in Section 6.4, requires 3.1 kilowatts output from the roving-vehicle fuel cells for 40 minutes, with an associated fuel penalty of 1.5 pounds of hydrogen and oxygen. A weight summary for the power equipment chargeable to the payloader hydraulic actuation system is as follows.

	Pounds
Hydrogen (includes preheating)	23.0
Oxygen (includes preheating)	20.0
Hydrogen tank	48.0
Oxygen tank	17.0
Gearbox	6.0
Speed control with tachometer generator	7.2
Turbine	10.8
Combustor	8.0
Radiator (including fluid)	88.0
Tubing, insulation, valves, structure	25.0
Heat-transport fluid and pump	5.0
Total	258.0

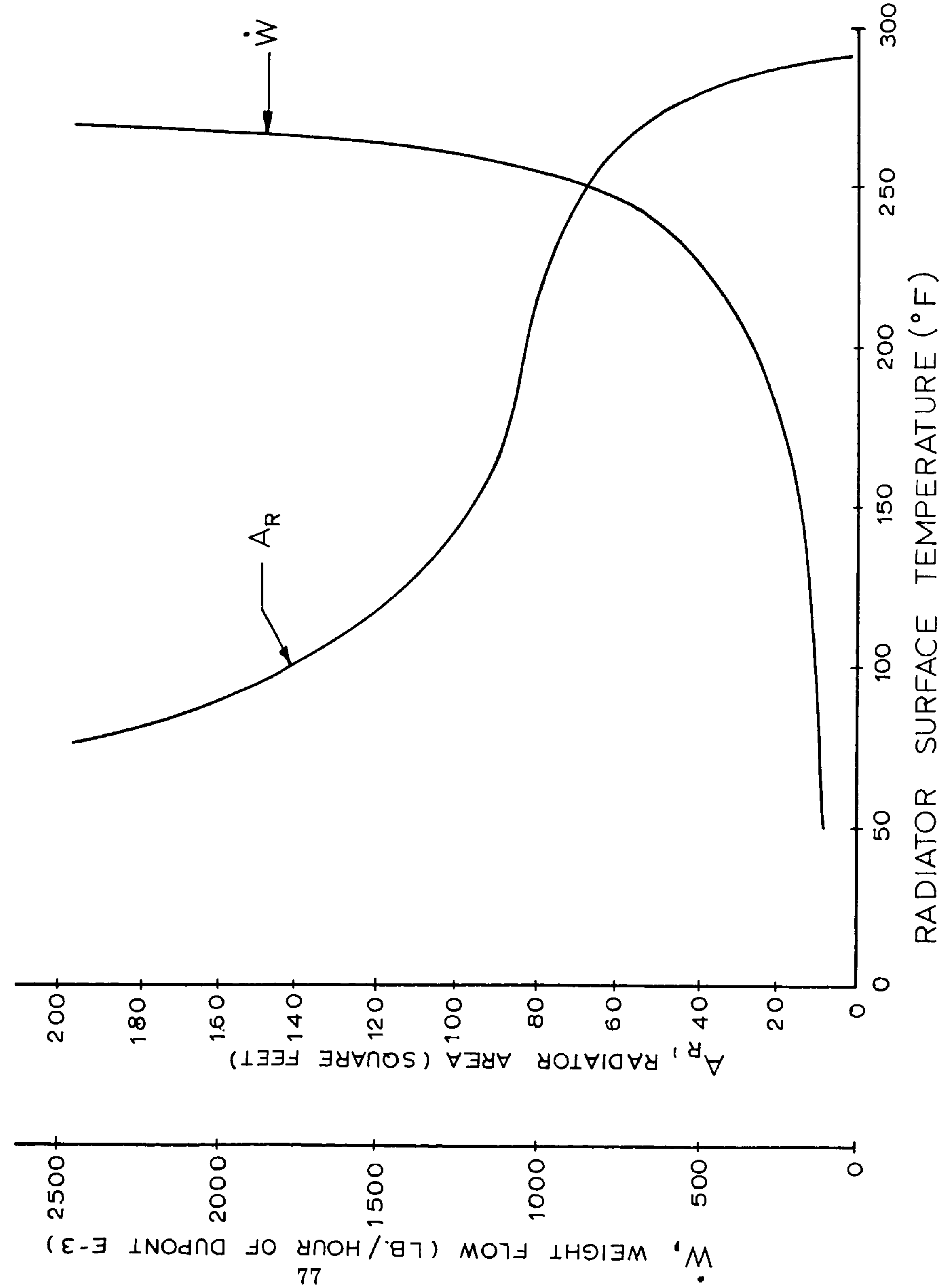


Figure 20: Radiator Size For Heat Rejection To Lunar Surface

7.2.2 POWER FOR ELECTROMECHANICAL CONCEPT

The accessory power unit needed to satisfy the electromechanical actuation of the payloader provides power for both the boom and bucket actuation functions. The power-load profile to perform these functions is shown in Figure 21. The turbine power unit design is similar to the unit previously discussed for the hydraulic concept. The power-unit for the electromechanical concept incorporates a voltage regulator in place of the tachometer generator used with the hydraulic power unit, as shown on the system schematic in Figure 21.

The average electromechanical load at the alternator is 3.72 horsepower provided by a turbine delivering an average of 9.3 horsepower and a maximum of 21.1 horsepower. APU fuel consumption is 10.9 pounds per hour at an H_2 to O_2 ratio of 1.17.

The electromechanical alternator losses average 3160 Btu/hr and are absorbed in preheating the hydrogen for the turbine. The hydrogen heat capacity is 9200 Btu/hr to provide a turbine-inlet temperature of 460°R. Similar to the design for the hydraulic power unit, the alternator losses are supplemented by the turbine exhaust to heat the hydrogen, and the gearbox losses (averaging 7400 Btu/hr) are rejected by a radiator. A radiator having 70 square feet of area is to be used for the hydraulic power unit. Preheating at minimum lunar-surface temperature is also required with the power system for electromechanical actuation. This is provided at a fuel penalty of 1.4 pounds of hydrogen and oxygen.

A weight summary for the power equipment chargeable to the payloader electromechanical actuation system is as follows.

	Pounds
Hydrogen (includes preheating)	23.2
Oxygen	20.0
Hydrogen tank	49.0
Oxygen tank	17.0
Gearbox	8.0
Voltage regulator	5.0
Speed control	4.0
Turbine	12.3
Combustor	9.0
Radiator (including fluid)	88.0
Tubing, insulation, valves, structure	28.5
Heat-transport fluid and pump	5.0
Total	269.0

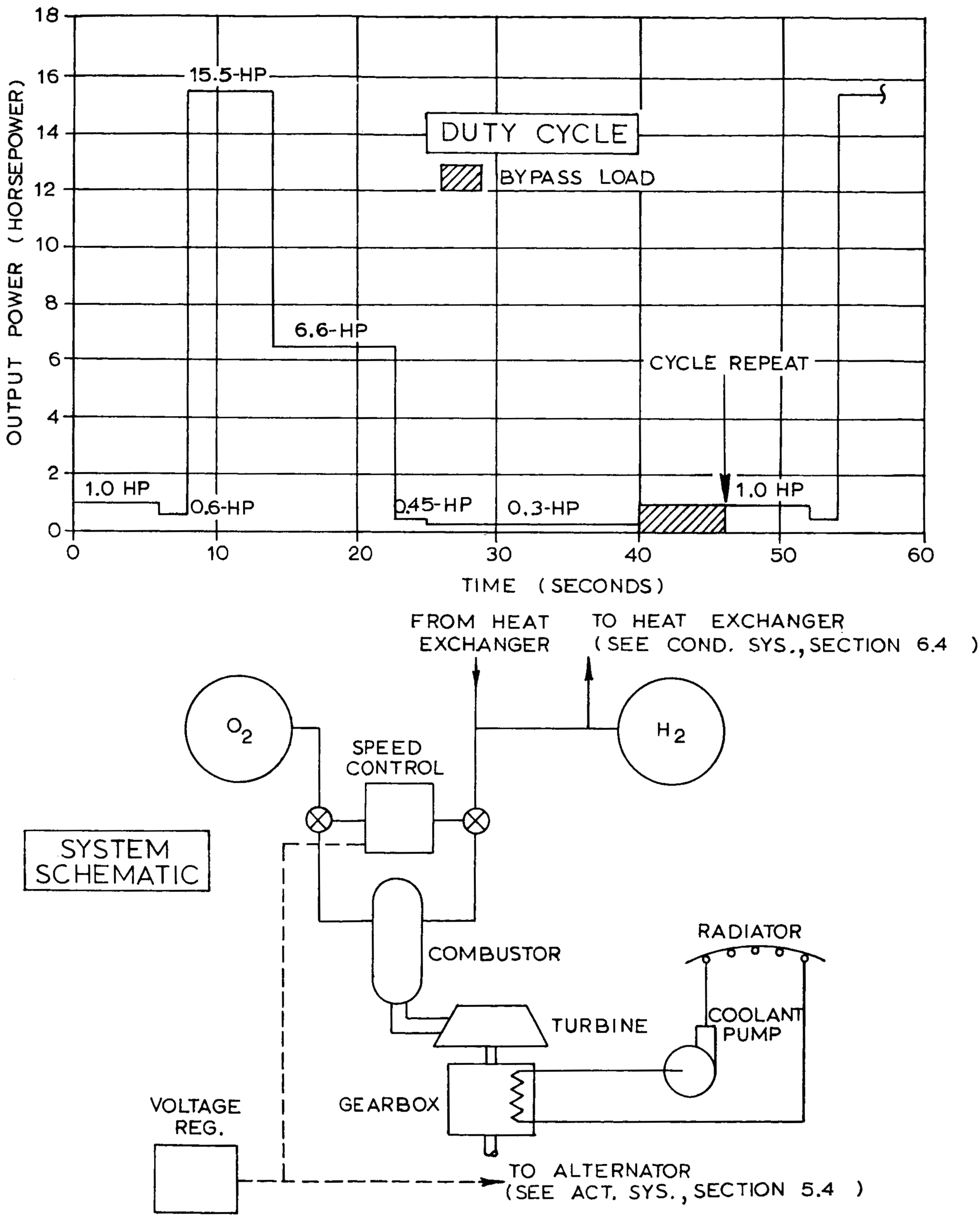


Figure 21: Payloader Actuation Power System Electromechanical Concept

8.0 COMPARATIVE EVALUATION

The comparison of actuation concepts for each of the four tasks evaluated is the concluding step in the actuation applications trade study. The comparative evaluation provides a method of reducing the important information from supplier surveys, task and requirement analyses, and system designs into numerical ratings of systems using evaluation parameters particularly important to space tasks. The parameters selected for this study are weight, reliability, cost, availability, and performance.

The use of weight as a trade parameter is particularly important to space vehicles in that the excess weight of accessory systems reduces the weight of deliverable payload by an equal amount. The booster penalty for additional payload weight is substantially greater than the weight added. The reliability of an actuation system on a space vehicle may determine the success or failure of the vehicle's mission. Particularly with flight control actuation equipment, the reliability may guarantee or imperil the safety of personnel aboard the vehicle. The measure of reliability includes consideration of the technical risks in using a particular system design approach, an evaluation of the complexity, and a measure of the loss of mission effectiveness if the system fails to operate.

The consideration of cost as a trade parameter is important in relation to the development of new hardware or to modification of off-the-shelf equipment. The availability of components and complete systems for space applications is measured by the lead time involved. The major consideration given to the analysis of availability is the extent of development required for each selected system and application. Equal performance is initially designed into all actuation systems for a given task; therefore, the comparative measure of performance is how well each system can maintain the designed level of performance with abnormal operational or environmental conditions.

A flow chart showing the sources of information and how they are used in the trade study is included in Figure 22.

A numerical method to compare one actuation concept to another for the same task was selected such that the method could be used for each of the four tasks evaluated. The numerical method allows reduction to percentiles of the numbers determined as representative of each actuation concept with respect to each of five evaluation parameters. The five percentiles, one for each parameter, are numerically combined to provide a final evaluation number representing a single actuation concept's acceptability for a given task. In all cases, the percentile numbers used are determined by a method such that no percentile used is either the highest possible (100 percent) or the least possible (0 percent). The evolution of the method selected for the numerical comparison is as follows.

An expression representing the position of each actuation concept evaluated, in respect to each parameter, was determined. As an explanatory example, it will be assumed that for some parameter an expression of X was determined to represent the hydraulic concept, Y for the pneumatic concept, and Z for the electromechanical concept. It is

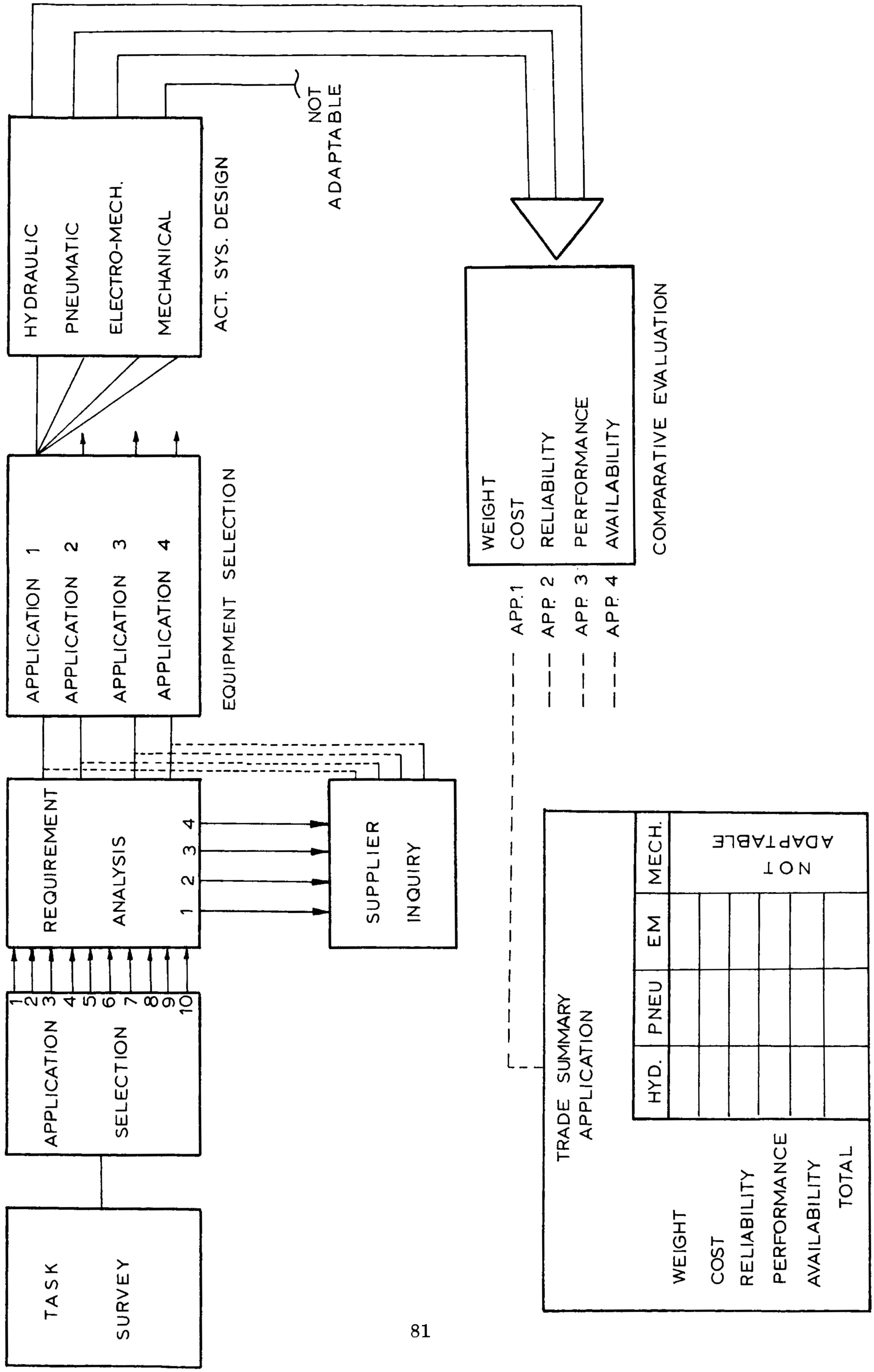


Figure 22: Trade-Study Flow Chart

observed that X has the least magnitude, Z the greatest magnitude, and that the greatest magnitude is the least desirable.

Based on the preliminary nature of the system designs for this study, it was concluded that some improvement in the evaluation expression representing a system design as related to any evaluation parameter should be expected when production hardware was available. To reflect this improvement capability, a 100-percent acceptable design in respect to any particular parameter was arbitrarily selected to be represented by an expression that is a 10-percent improvement over the best evaluation expression for that parameter in this study. Referring to the example cited, 100 percent is represented by the expression $X - X/10$. Determination of the expression representative of 0 percent is based on the criterion that if an evaluation were conducted for any actuation concept in relation to any parameter, that expression should be represented within the 0- to 100-percent scale. Using this criterion, it was not reasonable to determine the expression at 0 percent to be 2X, 3X, or some other multiplication factor.

Because the expression values and spread between values to be converted to percentiles reflect a large numerical range, and the magnitude of this range is not consistent from one task to another, the method to determine multiplication factors becomes complex. The method must fairly evaluate a case where only two closely competitive actuation concepts are adaptable, as well as a case where one of three concepts is clearly superior yet the remaining two concepts have specific advantages. It is concluded that an expression useful to selection of the multiplication factor or cutoff factor is the relationship of the most acceptable condition to the least acceptable condition. For the example, this expression is X/Z and is referred to as a guide number. The relationship between guide numbers and cutoff factors cannot necessarily be the same for all of the evaluation parameters. These relationships were determined by a "cut and try" method using guide numbers calculated for each parameter evaluation and each actuation task.

Further explanation is presented in the following weight analysis discussion.

8.1 WEIGHT ANALYSIS

The best approximation of the weight of each component in each system was made using vendor data, calculations based on the volume of material used, or comparisons of similar components. These weights are itemized in Section 5.0.

The summation of the component weights provides the total actuation system weight. The total weight chargeable to each system is this value in addition to the weight of conditioning equipment for the system plus the weight penalty to provide power for the system. These weights are shown in Figures 7, 8, 9, and 10.

Weight and guide numbers obtained in the study are as follows:

System	Elevon			Centrifuge		TVC			Payloader		
	Hyd	Pneu	EM	Hyd	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Weight	437	204	528	50	44	40	19	60	383	280	482
Guide Number	$\frac{204}{528} = 0.39$			$\frac{44}{50} = 0.88$		$\frac{19}{60} = 0.32$			$\frac{280}{482} = 0.58$		

In the specific example of the TVC system, the guide number (X/Z) = 19/60 = 0.32. Continuing with this example, the weight of the 100-percent system is (X - X/10) = 19 - 2 = 17 pounds. The cutoff factor for this system must be a number large enough so that the highest weight system will be within the 0- to 100-percent range. The minimum cutoff factor is, therefore, Z/(X - X/10) = 60/17 = 3.5. By the same calculations, the minimum cutoff factors for the other systems are 2.9 for the elevon (528/204 - 20), 1.25 for the centrifuge (50/44 - 4), and 1.9 for the payloader (482/280 - 28). Use of these factors will provide the least competitive system with a 0-percent evaluation.

To determine how much greater the cutoff factors should be above the minimum, the centrifuge and TVC analyses were examined, these having the highest and lowest guide numbers. The weights of the two adaptable centrifuge actuation concepts are close together, with only 6 pounds differential. Both systems are highly competitive; therefore, both should have representative percentile numbers. The weights of the TVC concepts are well spread, yet all three are low in magnitude, each concept having design merit. It was decided that the least desirable system should not be evaluated below approximately 25 percent.

The considerations stated above are graphically illustrated in Figure 23, which shows the development of the relationship between guide numbers and cutoff factors for the weight analysis. Using this curve, the number value representing the weight-zero percentile for each task may be determined as follows:

$$0\% = (\text{cutoff factor}) \left(X - \frac{X}{10}\right)$$

Using the TVC system as an example, the percentile ratings are determined as follows:

$$100\% \text{ equals } X - \frac{X}{10} = 19 - 2 = 17$$

Guide number = X/Z = 19/60 = 0.32

Cutoff factor using curve = 4.2

$$0\% \text{ equals (cutoff factor)} \left(X - \frac{X}{10}\right) = (4.2)\ (17) = 71$$

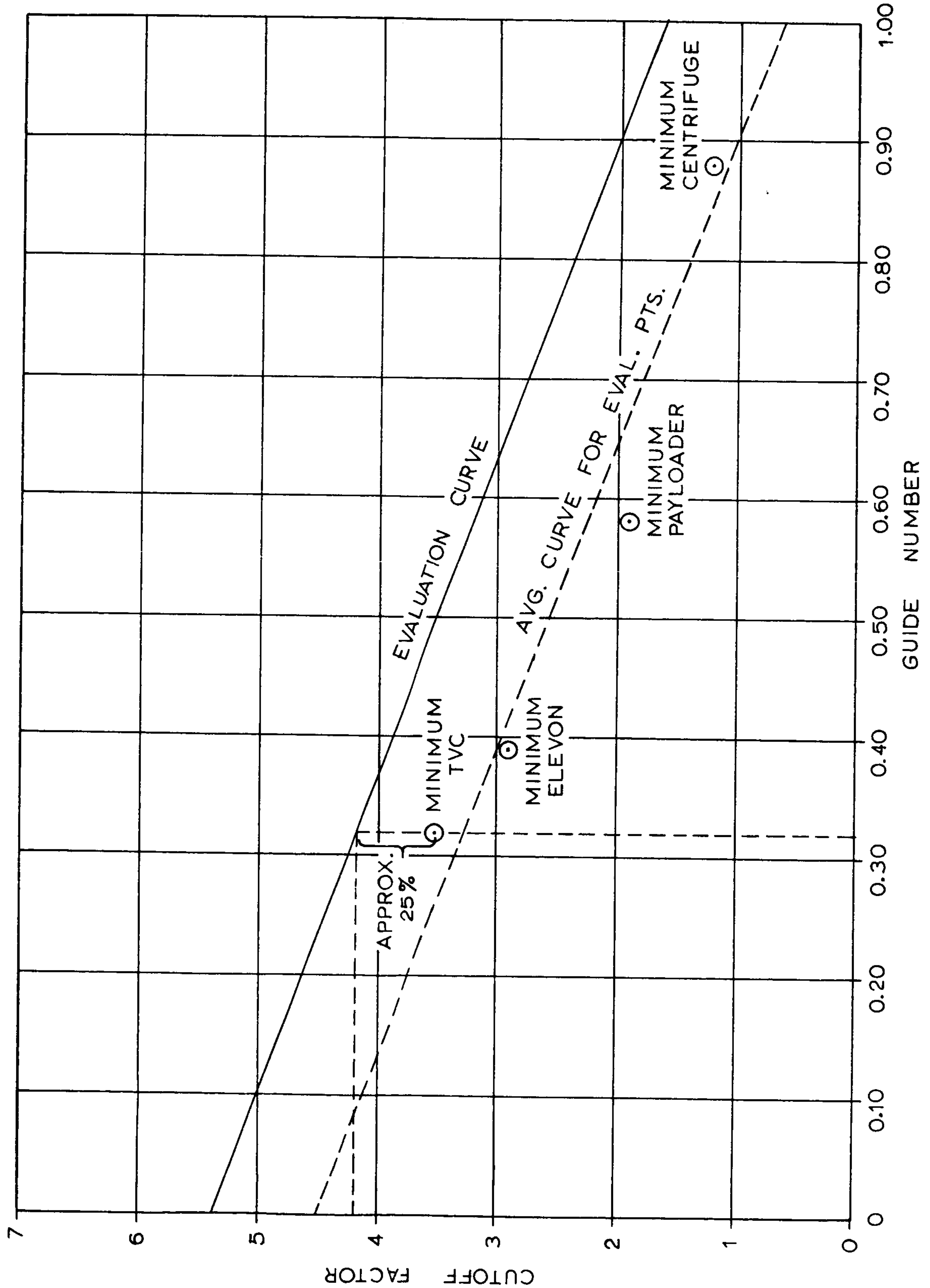
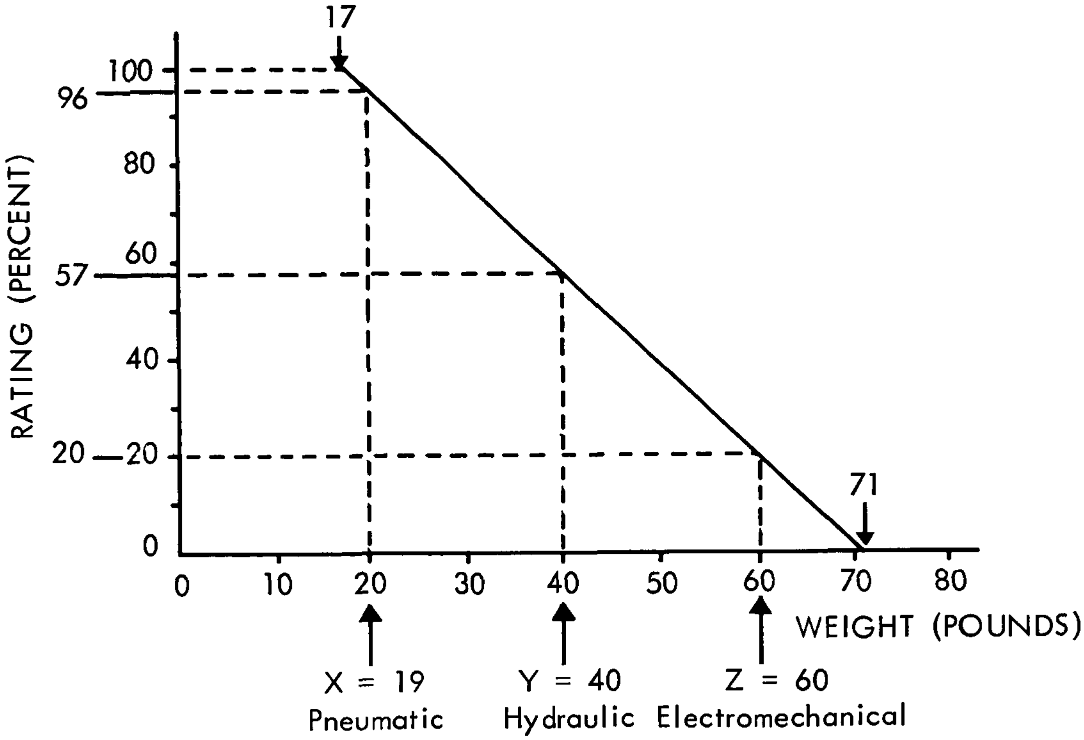


Figure 23: Weight Analysis Allowable Range

Using these values, the following curve can be constructed:



The percentile ratings are determined in like manner for each of the tasks with the results shown in the following table.

	Elevon			Centrifuge		TVC			Payloader		
System	Hyd	Pneu	EM	Hyd	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Weight	437	204	528	50	44	40	19	60	383	280	482
Guide Number	0.39			0.88		0.32			0.58		
Cutoff Factor	3.9			2		4.2			3.2		
Percent	53	96	36	75	90	57	96	20	76	95	59

The percentile results from the weight analysis and from each of the following parameter analyses are combined to provide a final rating number representing each actuation concept for each task. This method of combining these percentiles is described in Section 8.5.

8.2 RELIABILITY ANALYSIS

Reliability is difficult to assess for components that have not been developed or have not been tested in the usage conditions for which they are required. The method selected to evaluate reliability for this study was to indicate expected reliability as an expression of system complexity, mission abort failure possibility, and repair capability.

Each actuation system component type used in the system designs shown in Figures 7, 8, 9, and 10 was classified as to its complexity with a number within a range of 1 to 10. The ratings are as follows.

Complexity Rating	Item	Complexity Rating	Item
1	Tubing	3	Actuator
1	Filter	3	Brake
1	Solid Gas Generator	4	Flow Regulator
1	N_2H_4 Tank	4	Gearbox
1	N_2H_4 Reactor	4	Magnetic Clutch
2	Relief Valve	4	1/2 Spring Clutch
2	Check Valve	5	Four-Way Solenoid
2	Solid Igniter	5	Pneumatic Switching Valve
2	Gear Train	5	Differential Piston Bucket Actuator
2	H_2O_2 Gas Generator	6	Power Hinge
2	Reservoir	6	Electrical Generator or Alternator
2	N_2H_4 Start Valves	6	Electrical Motor
2	Solenoid	7	Electrical Motor with Controller
3	Pressure Regulator	8	Servovalve — Flow Regulator
3	Reservoir/Accumulator Combination	9	Hydraulic and Pneumatic Motors
3	LH_2, LO_2 Tank with Heater	10	Hydraulic Pump

Each component was also given a number between 1 and 10 as an expression of how a failure of that component affects the ability of the system to complete the task. The numbers given are indicated by the following descriptions.

Repair-Failure Rating

	Condition	Component
1.	No effects	Fluid, gas
2.	Failure will cause abort only as second failure in series	First item of parts in redundant installations
3.	a) Failure will stop operation, but replacement is possible	All centrifuge components except gearbox or hinge

b)	Failure would cause abort, but likelihood of occurrence is very small	Tubing, wiring
4.	Failure causes minor performance reduction	Filter, reservoir/accumulator, check-valve, solid propellant, motor abort
5.	Failure causes major performance reduction	Relief valve, actuator, first bucket hinge on payloader
6.	Abort failure of minor component having one condition of failure	Electromechanical gears
7.	Abort failure of major component with one condition of failure	Centrifuge gearbox or hinge, second bucket hinge payloader, payloader external gearbox, payloader elevon hinge, TVC ball screw, solid-propellant generator maneuver
8.	Abort failure of major component with two conditions of failure	TVC and elevon electromechanical motors, magnetic clutch, 1/2 spring clutch
9.	Abort failure of major component with three conditions of failure	Servovalve, hydraulic pump
10.	Failure gives no warning of immediate abort	TVC input gearbox hydraulic, TVC pneumatic regulator, elevon power hinge

Because the reliability of one component affects that of another, the system complexity is the product of the numbers representing each included component. Redundant components are expressed by a "product over sum" expression from the individual component numbers. The number to represent the repair-failure reliability for the system is also determined in the same manner as the complexity number. The combination of these numbers represents the unreliability of the system since the number becomes larger with decreasing reliability. The product of the system "complexity" number and the system "repair-failure" number is the numerical expression for total reliability.

The reliability rating for the TVC hydraulic system design shown in Figure 9 is presented as an example.

$$\text{Complexity Rating} = (3)\left(\frac{8 \times 8}{8 + 8}\right)(1)\ (2)\ (3)\left(\frac{8 \times 8}{8 + 8}\right)(3)\left(\frac{10 \times 10}{10 + 10}\right)(4)\ (1) = 9.8 \times 10^3$$

$$\text{Failure-Repair Rating} = (5)\left(\frac{2 \times 9}{2 + 9}\right)(4)\ (5)\ (4)\left(\frac{2 \times 9}{2 + 9}\right)(5)\left(\frac{2 \times 9}{2 + 9}\right)(10)\ (3) = 8.7 \times 10^4$$

$$\text{Rating Total} = (9.8 \times 10^3)\ (8.7 \times 10^4) = 8.5 \times 10^8$$

Ratings calculated for each system design in each evaluated task were reduced to percentiles by the methods previously discussed as part of Section 8.1. Because of the large size of the reliability numbers to be reduced to percentiles, only the nearest exponent of 10, as related to each calculated number, was used in determining the percentiles. The relationship of guide numbers to cutoff factors is illustrated in Figure 24. A summary of the results of the percentile calculations for reliability follows.

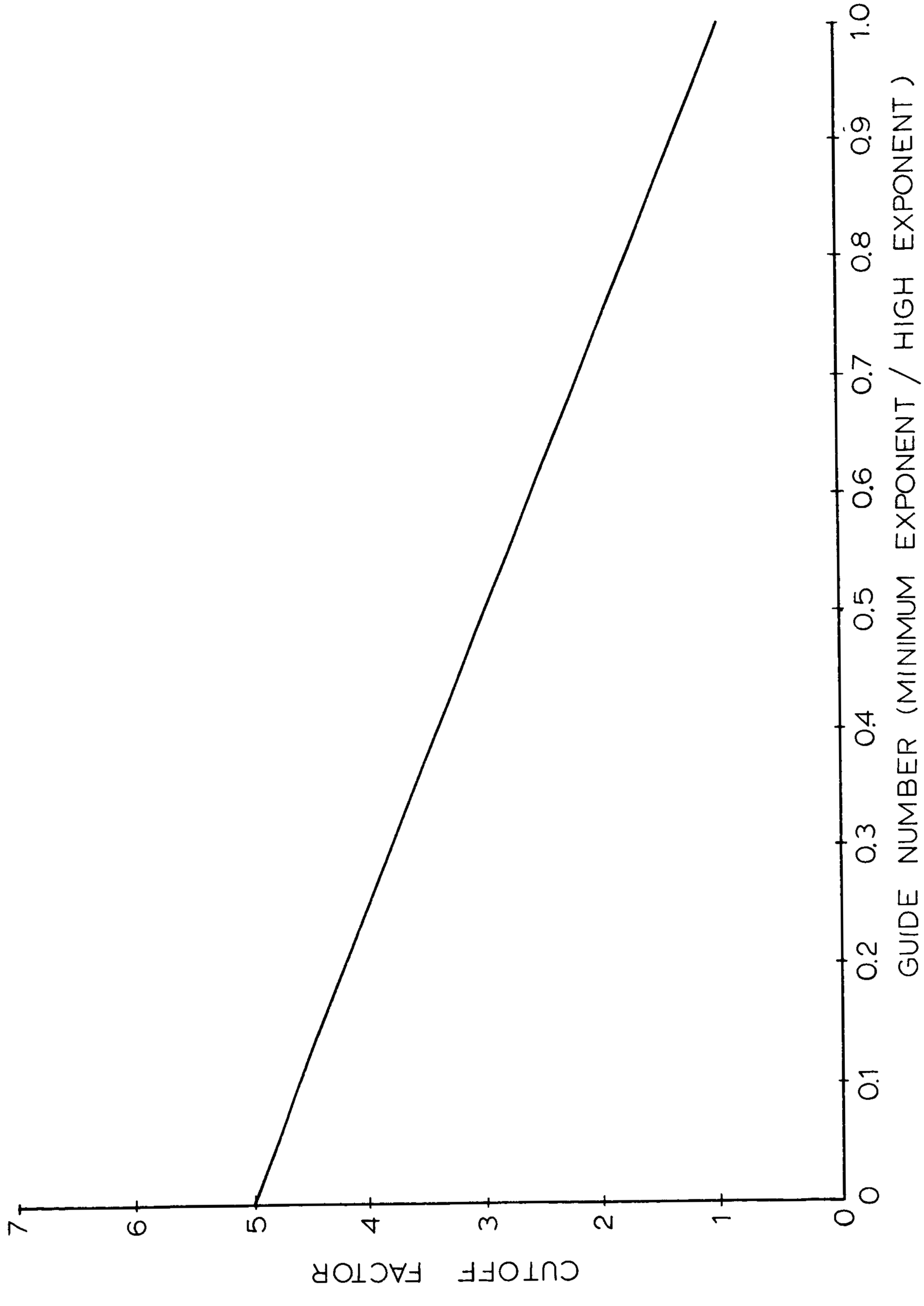


Figure 24: Reliability Analysis Allowable Range

System	Elevon			Centrifuge		TVC			Payloader		
	Hyd	Pneu	EM	Hyd	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Rating Number	6.2 x 10^8	9.5 x 10^9	1.1 x 10^{10}	3.5 x 10^9	3.3 x 10^5	8.5 x 10^8	5.8 x 10^5	6.7 x 10^8	1.8 x 10^{23}	1.0 x 10^{26}	7.8 x 10^{16}
Rating Exponent	9	10	10	9	5	9	6	9	23	26	17
Guide Number	$\frac{9}{10}$ = 0.9			$\frac{5}{9}$ = 0.56		$\frac{6}{9}$ = 0.67			$\frac{17}{26}$ = 0.65		
Cutoff Factor	1.4			2.8		2.3			2.4		
Percent	72	41	41	44	94	49	91	49	64	50	92

8.3 COST ANALYSIS AND AVAILABILITY ANALYSIS

The majority of the costs for actuation systems and components, as described in Section 5.0, is in development. The length of development time is a measure of availability which may, in certain instances, be shortened by increasing costs. From these statements, it may be seen that cost and availability are mutually dependent.

Accurate cost and availability analyses depend on obtaining data from suppliers with reasonable knowledge of the development problems with advanced space systems and the associated environments. Because suppliers, particularly in the hydraulic and pneumatic industries, have not pursued development toward low-temperature environments, accurate cost and availability data is not presently available. To provide a reasonable analysis relating costs and availability for the trade study, a list of components in the order of expected increasing costs was developed and is shown in Figure 25. A similar list made in the order of expected length of required development time is shown in Figure 26.

To determine the cost of a given system, the cost numbers for the individual components were summed. Each redundant or duplicate component in a given system received a value of 1, suggesting that the second article would not carry development costs. System cost numbers were reduced to percentiles by the methods previously discussed as part of Section 8.1. The relationship between guide numbers and cutoff factors is shown in Figure 27.

A summary of the results of the percentile calculations for the cost analysis follows.

System	Elevon			Centrifuge		TVC			Payloader		
	Hyd	Pneu	EM	Hyd	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Cost Expression	33	56	30	39	22	44	26	40	92	100	69
Guide Number	0.54			0.56		0.59			0.69		
Cutoff Factor	2.5			2.5		2.4			2.0		
Percent	85	29	93	44	94	34	91	47	52	39	92

Numerical Value	Elevon	Centrifuge	TVC	Payloader
1	Hydraulic Pneumatic Filters Hydraulic Relief Hydraulic Tubing	Hydraulic Filter Hydraulic Relief Hydraulic Tubing	Hydraulic Pneumatic Tubing Pneumatic Filter	Pneumatic Start Valve Hydraulic Pneumatic Tubing Hydraulic Relief Pneumatic 2-Way Solenoid
2	Pneumatic Tubing Pneumatic Check Valve	Hydraulic Packaging EM Packaging	Hydraulic Relief Hydraulic Filter EM Gear Train EM Ball Screw	Hydraulic Pneumatic Filter Hydraulic Pneumatic 4-Way Solenoid Pneumatic Pressure Regulator
3	Hydraulic Servo Pneumatic Relief EM Gear Train EM Packaging	Hydraulic A.C. Motor EM Motor/Brake	Hydraulic Packaging Pneumatic Packaging EM Packaging Pneumatic Pressure Regulator	Hydraulic Reservoir Pneumatic Combustion Chamber Pneumatic Reactor
4	Hydraulic Electrical Motor Hydraulic Actuator EM Motor Hydraulic Reservoir/Accumulator Hydraulic Pump	Hydraulic Servo	Hydraulic Servo Hydraulic Reservoir/Accumulator	Pneumatic Flow Regulator EM Packaging Pneumatic N_2H_4 Tank Pneumatic Switching Valve EM Gear Train
5	Hydraulic Packaging EM Spring Clutch	Hydraulic Pump Hydraulic Motor	Hydraulic Actuator	Hydraulic Electrical Motor Pneumatic Actuator EM Motor EM Alternator EM Motor
6	Pneumatic Actuator	EM Speed Control	Pneumatic Actuator	Hydraulic Motor Hydraulic Packaging Pneumatic O_2 Tank
7	Pneumatic Packaging EM Hinge	Hydraulic Reservoir	Pneumatic Servo	Hydraulic Bucket Pump Pneumatic H_2 Tank Pneumatic Motor
8			Hydraulic Gearbox EM Particle Clutch EM Motor	Hydraulic EM Bucket Hinge Pneumatic Packaging
9	Pneumatic Solid Rocket			Hydraulic Pneumatic Extension Gear Hydraulic Pneumatic EM Hinge EM Spring Clutch
10	Pneumatic Servo	Hydraulic Gearbox EM Hinge	Hydraulic Pump	Hydraulic Pump (Extension/ Elevation)

Figure 25: Component Cost

Numer-ical Value	Elevon	Centrifuge	TVC	Payloader
1	Hydraulic Filter Hydraulic Relief Hydraulic Tubing	Hydraulic Tubing	Pneumatic Filter Hydraulic Pneumatic Tubing	Hydraulic Pneumatic Tubing Pneumatic Start Valve Hydraulic Pneumatic 4-Way Solenoid, Hydraulic Pneumatic Filter Hydraulic Relief Pneumatic 2-Way Solenoid
2	Pneumatic Relief Pneumatic Filter Pneumatic Check Valve Pneumatic Tubing	Hydraulic Filter Hydraulic Relief	Hydraulic Relief Hydraulic Filter Pneumatic Regulator	Pneumatic Pressure Regulator
3	EM Assembly EM Gear	Hydraulic Electrical Motor Hydraulic Assembly EM Assembly	EM Gear Train Pneumatic Assembly	Hydraulic Bucket A.C. Motor EM Gear Train Pneumatic N_2H_4 Tank EM Assembly
4	Hydraulic Electrical Motor Hydraulic Reservoir/Accum-ulator Hydraulic Actuator	Hydraulic Servo	EM Assembly Hydraulic Assembly	Pneumatic Switching Valve Hydraulic EM Bucket Hinge
5	Hydraulic Pump Hydraulic Servo Hydraulic Assembly	EM Motor/Brake	EM Ball Screw	Hydraulic Motor Hydraulic Reservoir Pneumatic Combustion Chamber Hydraulic Assembly Pneumatic Reactor
6	EM Motor	Hydraulic Reservoir	Hydraulic Actuator	Pneumatic H_2 Tank Pneumatic O_2 Tank Hydraulic Bucket Pump
7	Pneumatic Assembly	Hydraulic Motor	Pneumatic Actuator Hydraulic Servo	EM Motor Pneumatic Motor Pneumatic Bucket Actuator Pneumatic Assembly
8	EM Spring Clutch	Hydraulic Pump	Pneumatic Servo Hydraulic Reservoir/Accumulator EM Particle Clutch	Pneumatic Flow Regulator EM Alternator
9	EM Hinge Pneumatic Actuator	EM Speed Control	EM Motor Hydraulic Gearbox	Hydraulic Pump EM Spring Clutch
10	Pneumatic Rocket Pneumatic Servo	EM Hinge Hydraulic Gearbox	Hydraulic Pump	Hydraulic Pneumatic EM Hinge Hydraulic Pneumatic Gearbox

Figure 26: Component Availability

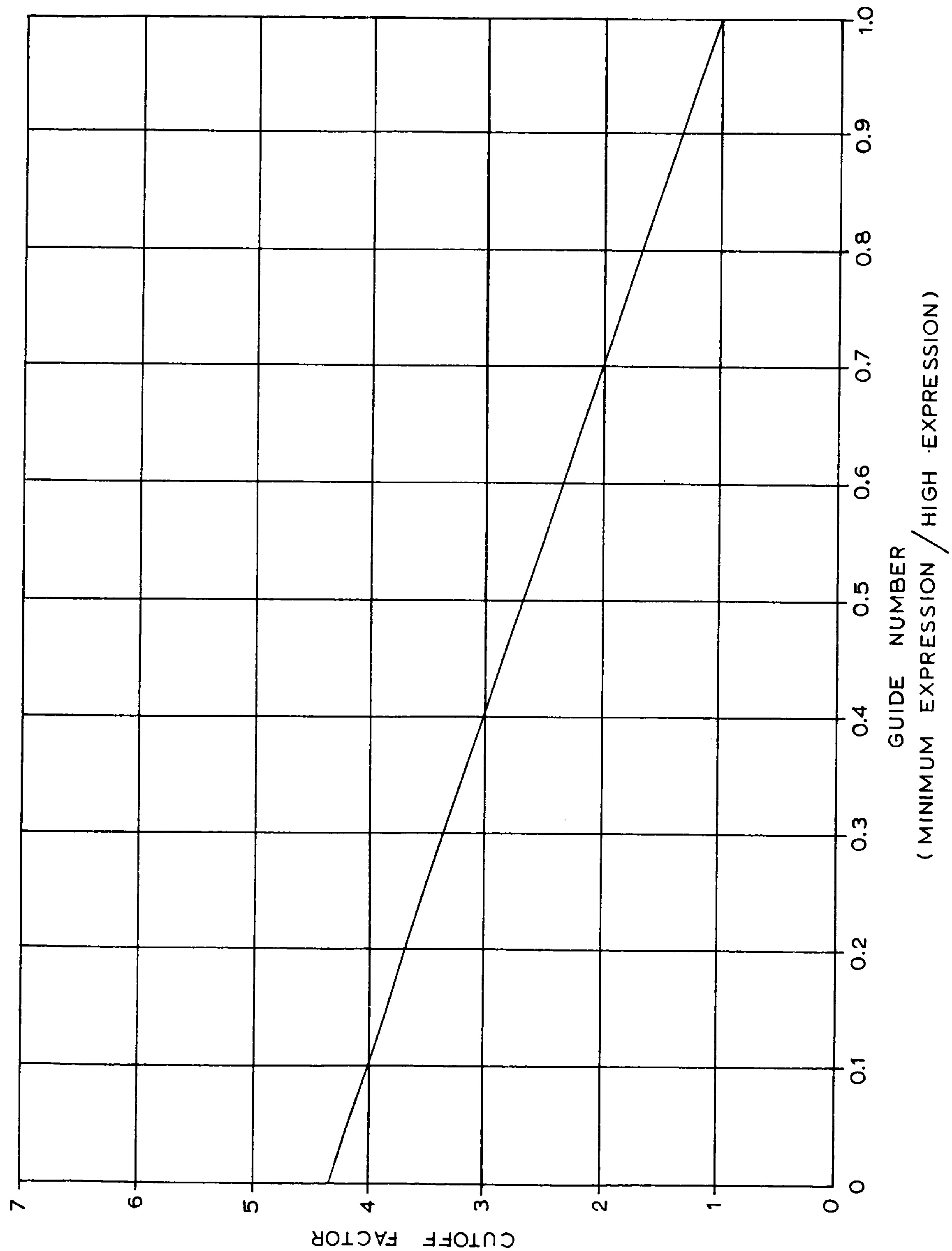


Figure 27: Cost, Availability, And Performance Analysis Allowable Ranges

The percentile ratings for the availability analysis were calculated in a manner identical to that used to obtain the cost percentiles. The relationship for guide numbers and cutoff factors is illustrated in Figure 27. A summary of the results of the percentile calculations for the availability analysis follows.

System	Elevon			Centrifuge		TVC			Payloader		
	Hyd	Pneu	EM	Hyd	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Availability Expression	36	61	37	46	28	54	27	47	81	110	71
Guide Number	0.59			0.61		0.50			0.65		
Cutoff Factor	2.4			2.3		2.7			2.2		
Percent	91	36	89	36	91	27	93	44	78	40	91

8.4 PERFORMANCE ANALYSIS

The selection of criteria that show performance differences, depending on the concept selected for a given task, is somewhat limited. In this study, all system concepts designed to perform a given task were also designed to give the same performance. Performance differences among systems may only be estimated in relation to requirement changes not expected at hardware conception.

The performance analysis was accomplished by listing all of the actuation components in the relationship to how performance would generally be affected by:

1) Changes in length of operating time of the system;

2) Changes in the hinge moments for the actuation task;

3) Changes in length of exposure to space environments or severity of these environments.

This listing is shown in Figure 28 with numbers from 1 to 10 assigned to each component, the higher numbers reflecting the more severe performance decrease.

A numerical expression of the capability of each system to adjust to different requirements is obtained by summing the values for individual components in that system. Only values for the primary flow circuit are added, since use of redundant items will yield the same performance as the primary components.

The method of conversion of these numbers to percentages for the trade study is identical to that used in the cost analysis. A summary of the results of the performance analysis is as follows.

Numerical Value	Elevon	Centrifuge	TVC	Payloader
1	Pneumatic Check Valve Hydraulic Pneumatic Tubing	Hydraulic Filter Hydraulic Tubing	Hydraulic Relief Hydraulic Pneumatic Tubing	Hydraulic Pneumatic Filter Pneumatic Start Valve Pneumatic 2-Way Solenoid Hydraulic Relief Hydraulic Pneumatic Tubing
2	Hydraulic Pneumatic Relief Hydraulic Pneumatic Filter	Hydraulic Reservoir Hydraulic Relief	Hydraulic Pneumatic Filter Hydraulic Reservoir/Accumulator Hydraulic Gearbox	Hydraulic Fluid Hydraulic Pneumatic 4-Way Solenoid Hydraulic Reservoir Hydraulic Pneumatic Gearbox Hydraulic Pneumatic EM Hinge EM Gear Train Pneumatic Bucket Actuator Pneumatic Switching Valve Pneumatic N_2H_4 Tank
3	Hydraulic Servo Hydraulic Electrical Motor EM Gear Pneumatic Actuator Hydraulic Reservoir/Accumulator	Hydraulic Gearbox EM Hinge	Pneumatic Pressure Regulator EM Gear Train	Pneumatic Pressure Regulator EM Spring Clutch Hydraulic Pneumatic Motors
4	Hydraulic Fluid EM Hinge Hydraulic Actuator Pneumatic Servo EM Spring Clutch	Hydraulic Fluid Hydraulic Electrical Motor	Hydraulic Fluid	EM Alternator Hydraulic Electrical Motor
5		EM Speed Control	EM Ball Screw	
6		Hydraulic Motor	Pneumatic Actuator	Pneumatic H_2 Tank Pneumatic O_2 Tank
7	EM Motor	Hydraulic Servo		
8	Hydraulic Pump		Hydraulic Actuator Hydraulic Servo	Pneumatic Flow Regulator
9			EM Particle Clutch Pneumatic Servo EM Motor	
10	Pneumatic Rocket	Hydraulic Pump EM Motor/Brake	Hydraulic Pump	Hydraulic Pump EM Motor Pneumatic Combustion Chamber Pneumatic Reactor

Figure 28: Component Performance

System	Elevon			Centrifuge		TVC			Payloader		
	Hyd	Pneu	EM	Hyd	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Performance Expression	48	43	36	40	21	54	36	76	58	96	43
Guide Number	0.75			0.52		0.47			0.45		
Cutoff Factor	1.8			2.6		2.8			2.8		
Percent	38	58	85	30	93	62	93	24	73	19	94

8.5 EVALUATION SUMMARY

The results of the analyses for the five trade parameters are brought together to formulate an overall rating of the competitive systems for each given task. The five parameters do not carry equal emphasis on the rating of a system for a given task. The importance of each evaluation parameter varies with the nature of the task considered. The most distinctive example of this is that performance is much more important to flight control tasks than it is to actuation tasks like the centrifuge and payloader. The following table indicates the emphasis given to each parameter for each task. Numbers are assigned between 1 and 10, with 10 being assigned to the parameters having the most significance for a given task.

Emphasis Table

Actuation Task	Elevon	Centrifuge	TVC	Payloader
Weight emphasis	8	8	8	10
Reliability emphasis	10	10	10	8
Cost emphasis	6	6	5	5
Availability emphasis	5	4	4	6
Performance emphasis	8	4	6	4

The final rating number for each actuation concept as related to each task was determined by multiplying each of the five percentiles by the appropriate emphasis factors and summing the products to arrive at a single number. The numbers for the TVC evaluation are presented as an example. The largest number indicates the actuation concept most acceptable for the task evaluated.

TVC Lunar Landing Vehicle

	Hydraulic			Pneumatic			Electromechanical		
	Percent	Emphasis Factor	Product	Percent	Emphasis Factor	Product	Percent	Emphasis Factor	Product
Weight	0.57	8	4.6	0.96	8	7.7	0.20	8	1.6
Reliability	0.49	10	4.9	0.91	10	9.1	0.49	10	4.9
Cost	0.34	5	1.7	0.91	5	4.5	0.47	5	2.4
Availability	0.27	4	1.1	0.93	4	3.7	0.44	4	1.8
Performance	0.62	6	3.7	0.93	6	5.6	0.24	6	1.4
Final Rating Number			16.0			30.6			12.1
Most Acceptable System						X			

A summary of the product calculations and final rating numbers for all of the evaluated tasks is shown in Figure 29.

Trade Parameters	Elevon Re-entry Vehicle			Centrifuge Space Station			TVC Lunar Landing			Payloader Lunar Rover		
	Hyd	Pneu	EM	Hyd	Pneu	EM	Hyd	Pneu	EM	Hyd	Pneu	EM
Weight	4.2	7.7	2.9	6.0	Not Competitive	7.2	4.6	7.7	1.6	7.6	9.5	5.9
Reliability	7.2	4.1	4.1	4.4		9.4	4.9	9.1	4.9	5.1	4.0	7.4
Cost	5.1	1.7	5.6	2.6		5.6	1.7	4.5	2.4	2.6	2.0	4.6
Availability	4.6	1.8	4.5	1.4		3.6	1.1	3.7	1.8	4.7	2.4	5.5
Performance	3.0	4.6	6.8	1.2		3.7	3.7	5.6	1.4	2.9	0.8	3.8
Rating	24.1	19.9	23.9	15.6		29.5	16.0	30.6	12.1	22.9	18.7	27.2
System Selected	↑					↑		↑				↑

Figure 29: Trade Study Summary

9.0 TRADE STUDY CONCLUSIONS

The results of the trade study show that a hydraulic system has more advantage for use on the elevon control of the shuttle vehicle than does either a pneumatic or electromechanical system. The hydraulic concept of actuation is also shown to be very competitive for the payloader actuation task. The pneumatic concept of actuation has a distinct superiority in application to the lunar landing vehicle TVC task due to the insignificant penalty for gas power. Pneumatic systems did not compete as favorably on other tasks evaluated and was not competitive for the centrifuge actuation task. The electromechanical actuation principle has a distinctly superior application to the centrifuge actuation task and was slightly superior to other concepts for payloader actuation.

Caution should be exercised in comparing concept ratings between tasks; such as: the hydraulic elevon system at 26.9 is better than the hydraulic TVC system at 15.5. These numbers are not comparable. The only comparison that may be made with these numbers is between hydraulic, pneumatic, and electromechanical systems for the same task. The effects of the environmental control and power equipment chargeable to an actuation concept are not included in the trade study, except in the weight evaluation. These effects are not expected to change the overall conclusions of the trade study. The influence of improving the cost and availability analyses with supplier data is not expected to cause other than minor changes to the overall evaluation results.

The overall space actuation system trade study provides sufficient evidence to conclude that hydraulic, pneumatic, and electromechanical concepts all have application to space tasks. Hydraulic systems were shown to be very competitive on two of the four tasks evaluated. Because only four tasks from a possible 80 were evaluated, it is not immediately evident which concept should be selected for any of the other tasks. This conclusion suggests that trade studies should be performed for tasks as they are conceived. The trade study is only a guide to the most advantageous selection of an actuation system because other factors such as the convenience of available power or the severity of space environmental effects on the system may dictate system choice rather than the overall evaluation results.

The space environmental effects, particularly low temperature, must be seriously considered when selecting a space actuation system. Because the state of the art in low-temperature operation of actuation systems is not well defined, test evaluations are required to determine operational characteristics in this environment. Hydraulic systems are of particular interest from the standpoint of availability of low-temperature fluids and the effects of viscosity on system operation at low temperature.

Volume II of the contract report contains the evaluation results of tests conducted to evaluate a hydraulic system design that satisfies the design requirements of the TVC actuation system analyzed in the trade study. Tests were conducted to: determine the best available low-temperature fluid for high-pressure hydraulic use, determine system operational characteristics at temperatures from maximum lunar surface temperature through the low-temperature limit of the fluid, and investigate long-term exposure of hydraulic components in hard vacuum.

REFERENCES

1. D2-22921 — A Lifting Re-entry Horizontal Landing Type Logistic Spacecraft — Final Report, The Boeing Company, February 3, 1964 (Confidential).

2. D2-22606 — A Survey of Current Space Vehicle Control Systems, January 1963 Review, The Boeing Company, October 16, 1963 (Confidential).

3. D2-100006 — Lunar Excursion Module — Propulsion and Flight Dynamics, September 20, 1962 (Confidential).

4. C 402-SM-45767 — Manned Orbital Laboratory (Zero G) Saturn V, Class Mid-term Review, Douglas Aircraft Corporation, November 1, 1963.

5. Aviation Week and Space Technology, April 6, 1964, "Design Study Shows Zero G Space Station."

6. ASD-TDR-63-454 — Lunar Alightment Systems Investigation, June 1963.

7. D2-1000057-3B — Volume 3, Part 2, Initial Concept of Lunar Exploration Systems for Apollo — Final Report, The Boeing Company, November 15, 1963.

8. D2-1000057-3A — Volume 3, Part 1, Initial Concept of Lunar Exploration Systems for Apollo — Final Report, The Boeing Company, November 15, 1963.

9. D2-1000057-1 — Volume 1, Summary, Initial Concept of Lunar Exploration Systems for Apollo — Final Report, The Boeing Company, November 15, 1963.

10. ASD Technical Report 61-430 — Human Factors of Remote Handling in Advance Systems, Manned Ground Support Equipment, General Electric Company, September 1961.

11. 2272 D — RL10A3 Model Specification, Pratt and Whitney Aircraft, August 1962.

12. PD 439 — Power Hinge Designers Handbook, Curtiss-Wright Corporation.

13. PD 468 — Principles of Nonlinear Servo Operation, Curtiss-Wright Corporation, March 31, 1964.

14. REA 65-524 — Space Vehicle Actuation Systems Trade Study, Lear Siegler, Inc., Power Equipment Division, November 25, 1964.

15. A0914998E — Technical Proposal: Space Vehicle Actuation Systems Trade Study, Kearfott Division, General Precision, Inc., December 1964.

16. BPAD-864-15541R — Space Vehicle Actuation Systems Design Study, Bendix Products Aerospace Division, November 27, 1964.

17. 903A — Technical Proposal: Space Vehicle Actuation Systems, Study Report, Cadillac Gage Company, December 1964.